POWER

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Candid Chats

The Chief's Valedictory

"THIS is the last conference we shall have while I am in charge of this station, boys, as I have been asked to begin work next Monday as superintendent of power at the Steel Product Co.'s plant, over in Carbonville. I am mighty sorry to leave this city and to think that I shall see you boys less often; but the opportunity to take up this new work was one I could not in justice to myself refuse. Mr. Brown was good enough to tell me that he wished I could afford to stay here and that things have gone well in this plant for a long time; but I told him that you boys are an organization of which any company might be proud and that things are bound to go along smoothly with Bill Johnson here as your new chief. Mr. Brown told me that I might announce Bill's promotion today, and I do so with no end of satisfaction that the new boss of this station is going to be a man who has grown up with it, and not a stranger from outside. (Applause.)

"It looks kind of funny, perhaps, for me to say this about outsiders, for I am a stranger to the Steel Products employees, and when I get over there I shall have to work all the harder to make good and to win the confidence of the boys and let them see that I am their friend. Well, I do not know why I was picked out for the new work over there, unless it was on account of my interest in rebuilding boiler plants, and you know that we have had a lot of that kind of work over here which has given us all a line of experience that the fellows over in Carbonville have not been lucky enough to have. Anyway, the selection of a new chief is up to the management; only I am more than glad that Bill here, who is next in line, is to be my successor. Put it right there!

"These changes come with the years and have to be; but they are not easy. I'm not going to try to tell you boys what I think about your friendship; but if any of you get over to Carbonville and don't come to say 'Howdy' to me I shall be sorry enough. I have had plenty of reason to know how you fellows look at things since I have been on this job; and all I ask is that you give Bill the loyal support that I have had. It will not be all plain sailing for him, for the reason that you all know his ways of looking at things so well, and as long as he has been a subordinate, his opinions and yours have had plenty of opportunity to mix, and in a general sort of way one point of view has been as good as another in your mutual discussions. Now very likely Bill will issue instructions bye and bye that you fellows wouldn't put across if you were in his place; and you may find it a bit hard sometimes to obey them without a kick. That all comes out of our being so well acquainted. But just remember this: Bill is going to be responsible to Mr. Brown for this whole shooting-match, and so if some of his ideas are different from what yours are, or from those I have been backing up here, why just forget it, and let him take the blame, if there is any. I'll bet from what I know of Bill that the praise will be coming a big sight oftener than the blame, if you all buck up and do as he asks every time. I know him well enough, too, to know that he hasn't got the idea that he knows it all, and that he will be mighty glad to talk over things whenever he is at liberty. We have never had any frost on the chief's doorknob in this plant, and I'll wager there isn't going to be any closed season on straight talks with the new boss after I am gone.

"I shall always look back upon these seven years that I have worked with you with satisfaction, and I take this opportunity to thank you all for the help you have given me. If it hadn't been for Tommy Reynolds' pluck and good sense the night the auxiliary piping gave out between the main header and those duplex feed pumps we should have been shut down right on the peak, and who was it but little Henry Grogan that spotted the flaw in the conveyor chain just in time to keep the whole coal-handling system from tying itself in a double bow knot the day the directors were inspecting this station! Naturally they gave me more credit for the good record of this station than I deserved, but I know what's what and who's who in this establishment, and, believe me, Mr. Brown knows a lot more than we fellows down here think he does, too. Why, a man with one eye and that half blind can't read over this station log book and not realize that it has only been by team play that we've kept the cost of production down to 6 mills per kilowatt-hour for the past three years, and with just one shutdown, which lasted 9 min. Bill and I are going to exchange cost records so I shall know how things are going in that way, and I am going to try my best to beat your record over in my new place. It's going to be some job, though, when the new 5,000-kw. turbine gets down to business out there in the operating room. I wish everyone of you good luck, and if ever I can do anything for you be sure to let me know!"

(Contributed by H. S. KNOWLTON, *Cambridge, Mass.)*

Power Cost in Paper Mill

By Thomas Wilson

SYNOPSIS—Operating-power costs in uptodate electric-driven paper mill turning out over 150 tons of paper per day. Engine units are being replaced by bleeder turbines. A feature is a noncondensing geared turbo-generator operating against a back pressure of 50 lb.

On a site covering 16 acres the Richardson Paper Co., of Lockland, Ohio, has built two board mills, one felt mill and a box factory. Besides, there is an independent mill and power plant which will not be considered in the present article. The buildings are all of brick construction, mostly one story and basement, and cover about one-quarter of the site. The output of the two board mills averages close to 120 tons per day of 24 hr., of the felt mill, 30 tons, and 15 tons for the independent mill previously mentioned. During the present year a remodeled power plant will permit a considerable increase in the output of the board mills, at least 125 tons per day being expected, while the capacity of the other mills will remain about the same.

All the machinery is motor-driven and mostly group-driven by means of lineshafting and belting to the individual machines. Double-ended motors drive the beaters, one of the latter being belted to each side. Each paper machine is driven by two motors, one of the variable-speed type and the other a constant-speed machine. As usual, these motors are belted to lines of shafting which in turn drive the variable- and constant-speed elements of the board machine. Through a lineshaft and individual belting or gearing one motor drives six agitators and two stock pumps, while in the same way a single motor drives three jordans.

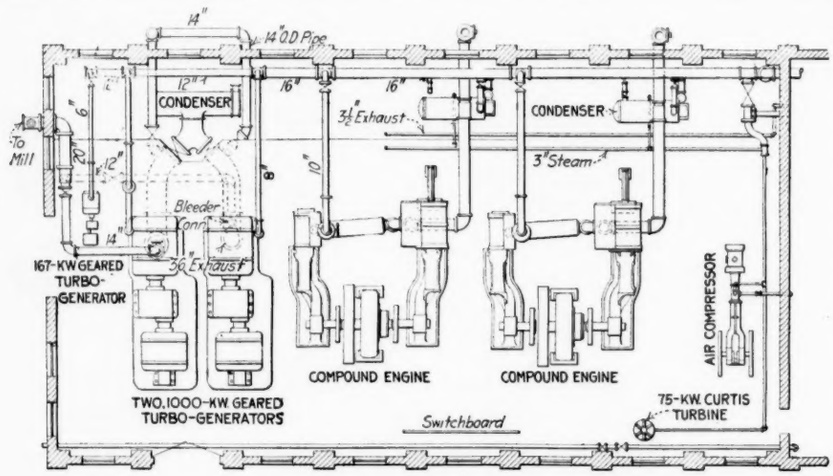

FIG. 1. PLAN OF THE ENGINE ROOM

There are approximately 100 motors in the mills, giving a total connected load of 3,500 hp. These motors are all loaded, and some of them are run at an overload at times. The electrical demand per day of 24 hr. varies from 40,000 to 50,000 kw.-hr. Of this about 700 kw.-hr. is required for lighting, which consists mostly of Mazda lamps connected two in series on 220-volt circuits. To provide this electrical energy and steam for use in the different mills, the following equipment has been in service in the power plant:

The boiler room contains 12 Stirling boilers (Fig. 2) having an aggregate capacity of 4,080 hp. and consisting of six 300-hp., two 364-hp., two 374-hp. and two 402-hp. units, all set in batteries of two. All these boilers are equipped with side-inclined stokers mounted in full-extension dutch-oven furnaces. The six smallest boilers, which were the first installed, are served by a brick stack 8 ft. in diameter and 170 ft. high above the boiler-room floor. Between the boilers and the stack are two economizers with a total surface of 12,000 sq.ft. To help the draft, a steel-plate fan is maintained in operation. The other boilers are served by steel stacks, one for each battery, which are 6 ft. in diameter and 170 ft. high. From 8 to 10 boilers are operated continuously, and the number served by the two economizers may be varied. The operating pressure is 155 lb. The steam is not superheated.

All condensation from the mills and the power plant is returned to a hotwell. The makeup amounting to 10 to 15 per cent., is condenser circulating water. Originally this water comes from the state canal, which flows past the plant, or from deep wells. It is passed through the condensers into a large storage tank, and from here as much of the supply as is needed is admitted to the hotwell under float control. It is then pumped through a vertical open heater and the economizers to the boilers. As the water leaves the heater, the temperature ranges from 160 to 210 deg., and in the economizer it ranges between 280 and 310 deg. In the entire plant 4,000,000 gal. of water per day is used, so that in addition to the storage tank just mentioned, a pond having a capacity of 1,500,000 gal. has been provided to insure at least a small reserve supply. For boiler feeding there are two duplex pumps, each having a capacity of 516 gal. per min., another somewhat larger pump and a fourth, very large, pump with cylinders 30x15x36-in., which is capable of supplying 5,500 b.hp. In addition there is a fire pump having a capacity of 1,000 gal. per min., also a low-service pump, which transfers the water from the hotwell to the heater. These pumps are all provided with governors, and at the boilers feed-water regulators control the supply.

West Virginia slack coal is burned in the furnaces. It comes in over a siding passing over the top of a large concrete storage bunker, the track level being considerably higher than that of the boiler-room floor. The coal is dumped from the cars directly into this bunker, and by means of individual scraper conveyors running from the bottom of this bunker, the coal is supplied to each stoker.

To supply the electrical energy two cross-compound engines, 23x46x36-in., directly connected to 220-volt two-wire direct-current generators have been operated at more than their rated capacity. Each is provided with a surface condenser having 1,800 sq.ft. of surface, but owing to the

overload on the engine and the high-temperature circulating water, the vacuum has not exceeded 24 in. and has averaged about 20 in. Under test conditions at this vacuum and the normal overload, the engine units developed a kilowatt-hour on 24 lb. of steam.

Circulating water for each condenser has been provided by a 10-in. fan pump driven by a 30-hp. steam turbine at 1,800 r.p.m. The wet-vacuum pumps were also steam-driven. The exhaust from these auxiliaries, in addition to that from a 75-kw. turbo-generator, is used for low-pressure drying in the mills. The latter unit has been operating in parallel with the engine-driven generators to help meet the demand on the plant and on Sunday, when the mills and the large generating units are shut down, is operated to furnish whatever lighting is required. This unit, as well as the turbines on the auxiliaries, operates against a back pressure of 15 lb. Provision is made to supplement this supply with live steam reduced to the pressure mentioned.

For certain high-pressure drying in the mills a temperature corresponding to a steam pressure of 50 lb. was found desirable. To meet this demand a 167-kw. turbine was installed to operate at this abnormal back pressure. It runs at a speed of 6,000 r.p.m. and is geared to a generator making 900 r.p.m. Any additional steam needed at this pressure is supplied directly from the boilers through reducing valves.

With such a large demand for exhaust steam in the mills, the desirability of operating the main generating units condensing may be questioned. In this case it was a question of securing all the power obtainable from existing units, and to operate cross-compound engines at the high back pressure required was not considered desirable.

Under the conditions just related the operating costs given in the accompanying tables were obtained. Table 1 gives the monthly expense and the total for the year 1915 for the various items that enter into the generating cost. It will be seen that the overhead costs, including interest on the money invested, depreciation, insurance, taxes and general office expenses apportioned to the plant, are included. There is also a rental item as the power plant is charged for the space occupied. In addition there are the usual items of wages for the engineering force, fuel, water, lubricants and waste, and supplies. There is a labor item for repairs and maintenance which is unusually large for a plant of the size, but this is due to remodeling the plant.

TABLE 1. POWER PLANT COSTS FOR YEAR 1915

	Per Month	Total for 1915
Wages, engineer and assistant	$2,310.99	$27,731.83
Fuel	6,683.59	80,203.02
Water charge	36.36	436.35
Lubricants and waste	64.11	769.35
Labor for repairs and maintenance	377.84	4,534.03
Supplies	865.24	10,382.86
Rental cost	589.73	7,076.76
Interest, depreciation, insurance and taxes	2,571.26	30,855.12
General office expenses, diffused	824.52	9,894.29
	$14,323.64	$171,883.61

TABLE 2. COSTS OF 1915 DISTRIBUTED BETWEEN THE VARIOUS MILLS

	Kw-Hr.	Cost	Coal	Labor
Board mill machine No. 1	4,447,024	$58,693.53	$27,408.21	$9,469.66
Board mill machine No. 2	3,577,906	47,277.31	22,101.86	7,610.89
Felt mill	4,651,694	61,501.53	28,638.84	9,939.62
Power-house lighting	281,381	3,748.76	1,744.03	605.50
Box factory	22,230	299.62	140.10	47.79
Mill No. 1 (light)	27,605	362.86	169.98	58.37
	13,007,840	$171,883.61	$80,203.02	$27,731.83

The data were obtained from monthly reports, such as is shown herewith. It will be noticed that daily records are kept of the usual operating data, the supplies used, amount of coal burned, repair items and boiler cleaning, as indicated by the headings. In addition to this, monthly record charts have been plotted giving the tonnage of paper in the different mills and the kilowatt-hours required.

Table 2 distributes the electrical output, total cost and the coal and labor items between the various mills. These costs were figured on an average charge per kilowatt-hour of 1.32c. It must be remembered that this charge includes overhead, a larger labor charge than would ordinarily be required and a continuous supply of live steam averaging 700 to 900 b.hp., which does not pass through the generating units and should not enter into the generating costs. In the month of October the cost per kilowatt-hour was 1.13c., or about 0.2c. less than the average for the year. Omitting the overhead expenses, the rental charge, two-thirds of the extra labor for repairs and maintenance and two-thirds of the supplies, much of the last two items being occasioned by remodeling of the plant, the operating cost per kilowatt-hour reduces

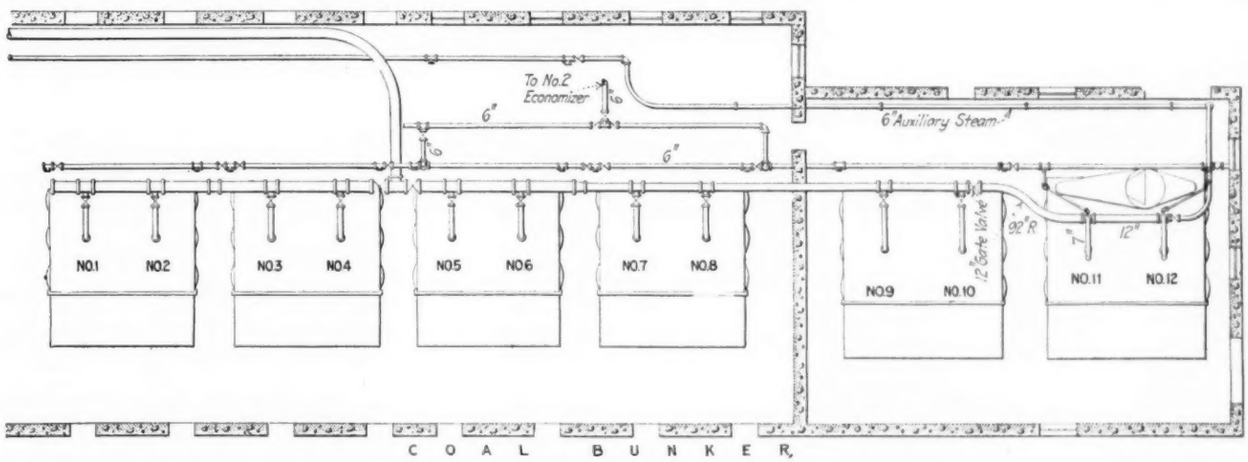

FIG. 2. PLAN OF BOILER ROOM, SHOWING MAIN PIPING

TABLE 3. A THREE-YEAR COMPARISON OF OUTPUT, COAL BURNED AND LABOR

	1915	1914	1913
Tons of paper, felt and board mills	37,044.11	31,457.68	32,426.00
Coal used, tons	48,805.55	41,580.45	41,134.26
Coal delivered, tons	48,528.55	40,169.55	43,391.15
Coal used per day, tons (364 days)	120.05	113.92	112.69
Coal used per kw.-hr., lb.	7.504	7.581	8.34
Coal used per ton of paper made, tons	1.3174	1.32179	1.27
Coal cost per ton of paper made	$2.1649	$2.2210	$2.1272
Coal, average cost per ton	1.64331	1.6803	1.675
Power-house labor per kw.-hr.	0.002132	0.002628	0.002919

to 0.88c., which is not bad considering the conditions and the supply of a large quantity of steam directly to the mills.

Table 3 gives some interesting figures on the amount of coal used during 1915 and the two years previous. In 1915 it will be noticed that 7½ lb. of coal was required per kilowatt-hour. This is of course high, but has been

Monthly Power House Report and Cost Sheet

...OF...

THE RICHARDSON PAPER CO.

Lockland, Ohio, ... 19..

DATE	K.W.H.	Eng. Units in Service	Eng. Hrs.	T. Units in Service	T. Hrs.	Total Hrs.	BOILERS IN SERVICE
1							L. P. Cylinder Oil on hand. ..... Gal.
2							" " " used ..... "
3							H. P. " on hand ..... "
4							" " used ..... "
5							Engine Oil on hand .....
6							" " used .....
7							Cylinder Oil to Mill. Credit P. H. ..... "
8							Eng. Oil to Mill. Credit P. H. ..... "
9							Cylinder Oil used per K. W. Hour
10							Eng. ..... "
11							Eng. Oil used per Eng. Hour ..... "
12							" " K. W. Hour .....
13							Engine Hrs. Turbine Hrs.
14							REPAIRS
15							
16							
17							
18							
19							
20							
21							
22							REMARKS
23							
24							
25							
26							
27							
28							
29							
30							
31							TOTALS
							DAILY AVERAGE

Week of ..... Started ..... Stopped .....

" " "

" " "

Power off during time needed, and cause

Dates and number of boilers cleaned

Condition of all power aux

Total cost of *all* labor to furnish power

" " " " per K. W. H.

" " Coal per K. W. H. as per tests of

" coal delivered

" " on hand (estimated)

accounted for previously. Per ton of paper made, an average of 1.3 tons of coal was burned, which must be judged by the quality of coal used and the conditions existing in the plant. Roughly, a pound of coal should produce a pound of paper when the power plant is modern and operated at the highest efficiency and when the demand for exhaust steam approximately balances the supply from the generating units.

To improve the operating conditions and to meet the growing demand for power, two 1,000-kw. bleeder turbines are being installed (Fig. 1). These machines, which operate at 3,600 r.p.m., are geared to generators that will have a speed of 600 r.p.m. A common condenser will serve the two units, and the auxiliaries will consist of a 12-in. motor-driven centrifugal circulating pump and an Edwards air pump. These units will be large enough to handle the entire load, but one of the engine units will be operated and carry just enough load to heat the circulating water through the condenser to 140 deg. It is planned to store this water in a large tank and recirculate it through the condenser so that it will be maintained at the proper temperature for the beaters, or at the 140 deg. previously mentioned. The circulating water for the turbine condenser will also be stored in a large tank supplying the mill. If the demand should be less than the amount of circulating water needed, the tank will overflow to the reservoir previously mentioned.

Sufficient steam will be bled from the turbines at 15-lb. pressure to supply the demand previously cared for by the 75-kw. turbine, the condenser auxiliaries serving the engine units, and live steam from the boilers. These same auxiliaries will be replaced by motor-driven pumps, and the 75-kw. turbine will be reserved for Sunday use to supply the lighting in the mills when the main generating units are out of service. The small geared turbine unit will be retained to supply steam at 50 lb. pressure until conditions in the mill are altered so that exhaust steam at moderate pressure will be suitable.

The new work is being installed under the supervision of W. E. Bodenstein, mechanical engineer for Stegner & Hughes, architects and engineers, of Cincinnati. Of the paper company J. M. Richardson is president; W. H. Richardson, vice-president; T. J. Mullen, general superintendent; F. A. Nusbaum, chief engineer of the power plant; and Gilbert Quackenbush, electrician in charge of all electrical work in the mills and the power plant.

Engineering Students Number Over Thirty Thousand

The appended tabulated figures on students in the engineering schools of the United States were presented at the recent annual meeting of the Society for the Promotion of Engineering Education, through the coöperation of the United States Bureau of Education and the society's Committee on Statistics. A variety of interesting comparisons are possible, such as the great increase in chemical engineers, the decrease in civil and mining, and the slump and recovery in number of mechanical engineers.

STUDENTS IN THE UNITED STATES

Engineering Branch	1906	1911	1915
General	2,501	5,591	6,059
Chemical	1,234	1,452	2,383
Civil	7,962	8,934	7,182
Electrical	5,696	6,128	6,637
Mechanical	7,426	7,052	7,751
Mining	2,826	2,337	1,922
Other	82	....	958
Total	27,727	31,499	32,843

Follower Bolts Lose Their Strength by crystallization caused by alternate cooling and heating and are likely to fracture when comparatively small strains are brought to bear. To be on the safe side, careful engineers make a practice of renewing these bolts at certain intervals.

Automatic Water-Works for a Small Town*

By Clyde Potts†

SYNOPSIS—In this plant the operation of the pumps is automatic, the electric motors being controlled by the water pressure. The control apparatus is so sensitive that the opening of a hydrant at any part of the pipe line will start the pumps.

The increase in the population of the borough of Milltown, N. J., from 560 in 1900 to 2,500 in 1912 found the town without water-works. Plans were drawn by the writer for a water-supply system. The following notes on the water-works should be of interest to engineers dealing with small-town problems, since the service is by pumping, but is automatic and requires only a small amount of attention.

The supply consists of a filter gallery laid along the edge of the hill and leading to a large storage well. In addition to the filter gallery two wells 6 ft. in diameter, made of interlocking steel sheet piling, were driven through the gravel to a shaly rock. These wells also flow into the storage well.

The filter gallery is some 500 ft. long, is 6 ft. wide and 6 ft. deep. The sides are built of hollow tile along the lower 4 ft. The groundwater enters the filter gallery through the sides and through the bottom, coming up through the gravel. This gallery has a reinforced-concrete roof and is provided with manholes at various points to permit the attendant to enter it for the purpose of cleaning. The storage well is 30 ft. in diameter and 18 ft. deep. Water is taken from this storage well by a Deane 10x10-in. triplex pump operated by an electric motor. A similar gas-engine unit is installed (Fig. 1). The water is pumped through a 10-in. pipe to a 90,000-gal. tank elevated on a 75-ft. tower.

The electric pump is automatic in its operation, the electric motor being controlled by a water-pressure gage on a special electric switchboard. The electric energy used is from the town lighting mains—two-phase alternating current at 60 cycles and 220 volts.

When the water in the standpipe falls below a certain point, the reduced pressure on the pressure gage at the pumping station closes a switch, which starts the electric pump in operation, and water is pumped from the storage well into the standpipe until it reaches a predetermined level, when the pressure gage cuts off the current at the switchboard and the pump stops. The control secured by this pressure gage is so delicate that the opening of a hydrant at any place in the main part of the borough reduces the pressure a sufficient amount so that the electric pump starts. In the event of a fire in the borough not only is the water in the standpipe available, but the opening of a hydrant by the firemen immediately starts the electric pump at the pumping station, and the 90,000 gal. of water stored in the storage well also becomes available through direct pumping.

The cost of the wells and supply was $6,200. The cost of the pumping station, including the electrically driven pump and gas-engine auxiliary, was $7,650; and of the standpipe and tower complete, $5,800. With the cost of the pipe system before mentioned, this makes the total construction cost of the water system $43,572.

The plant is automatic in its operation and requires very little attention. The superintendent of the water plant, who also has charge of the sewer system, visits the plant daily for purposes of inspection and oiling. The cost of pumping since the plant has been in operation, from Apr. 1, 1915, has been a trifle under $14 per million gallons, exclusive of superintendence.

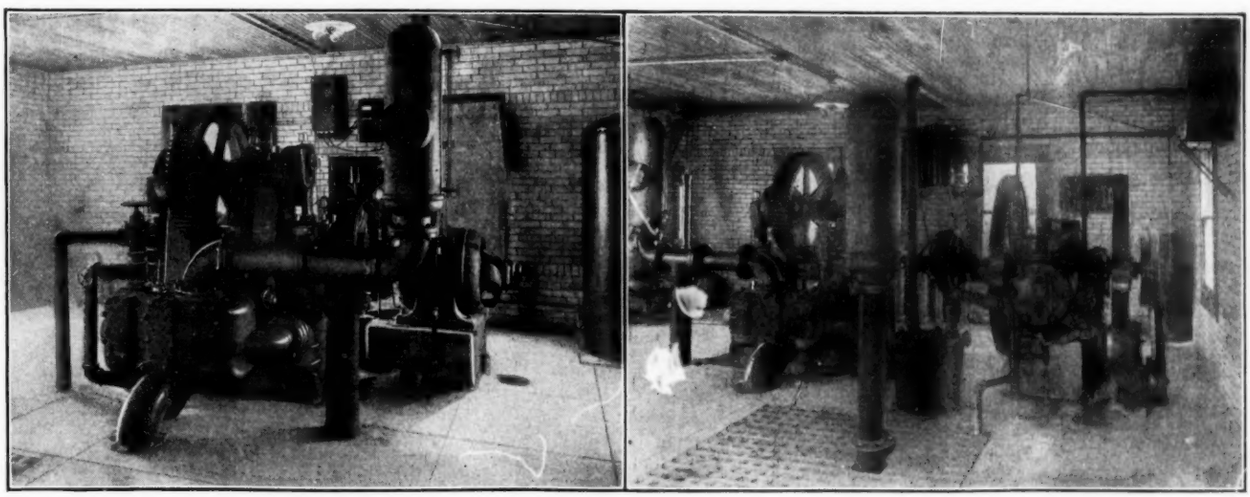

FIG. 1. INTERIOR OF THE MILLTOWN AUTOMATIC WATER-WORKS

Since the most unusual feature of this plant is its automatic starting and stopping device, a more detailed description of this controller may be of interest. It is known as the Whittingham motor-driven type, having a double-pole magnetic main-line switch, double-pole no-volt relay sylphon-pilot pressure regulator, three-pole single-throw 200-amp. knife switch and fuses, pressure

*"Engineering News," June 15, 1916.

†Consulting Engineer, 30 Church St., New York City.

gage and a switch to permit hand operation independently of the automatic control.

The device is distinctive in that, while it is of the current-limiting type and absolutely prevents the motor from being started on an overload, it also provides for cutting out the resistance if the motor starts and in this respect is unlike series current-limiting controller devices. The resistance, instead of being cut out as is customary on a comparatively few steps, is cut out in 24 graduated steps at an adjustable speed, which once set is unchangeable. This is obtained from the motor through a worm gear magnetically clutched to the resistance arm, resulting in absolute freedom from arcing and from changes in timing. The resistance is of a sand-packed tubular type, mounted below the line of panel, leaving all back wiring exposed. This back wiring is of fireproof insulation, with soldered lugs at all connecting points. The period of cutting out of the resistance, or reaching full speed, is considerably over 1 min. in this case, which is highly desirable on a triplex pumping engine of this size, where it is especially needed to keep down the inrush of current.

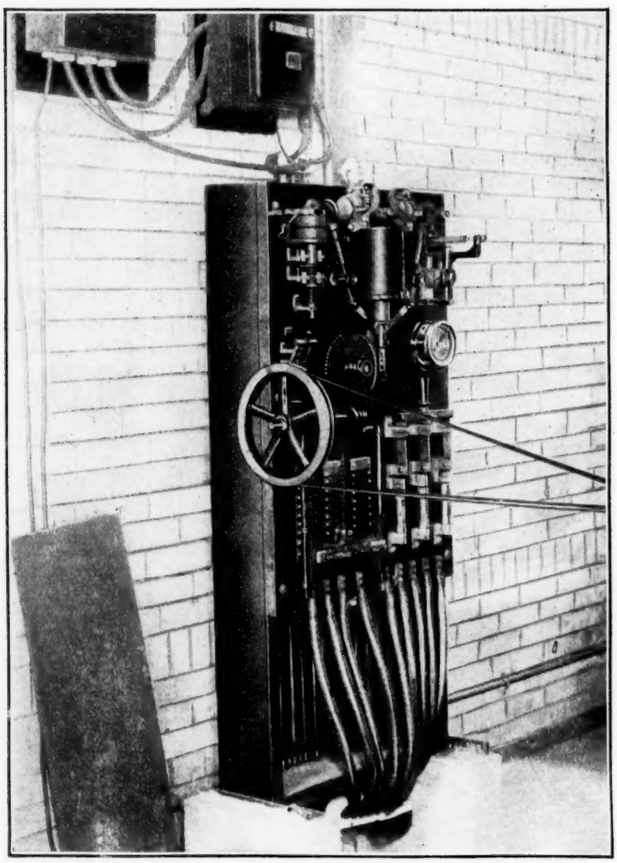

FIG. 2. AUTOMATIC ELECTRIC PRESSURE REGULATOR

At full motor speed but two magnets are energized, and this detail differs materially from most controlling devices for the purpose. There is freedom from noise and heating, such as are frequently encountered where a large number of solenoids are called upon to cut out the resistance in successive steps.

This controller not only provides protection against an overload, but also against the blowing out of a fuse, as the relay operated by the pilot-pressure regulator is across one phase, while the magnetic main-line switch is across the other. Therefore, it would be impossible to close the main-line switch, which is fed through the relay contacts, if either phase opens.

In service the regulator is intended to stop the pump at a certain predetermined level about 10 or 12 ft. above the low level at which it starts; but owing to the frictional head, it is found that the opening of a hydrant will

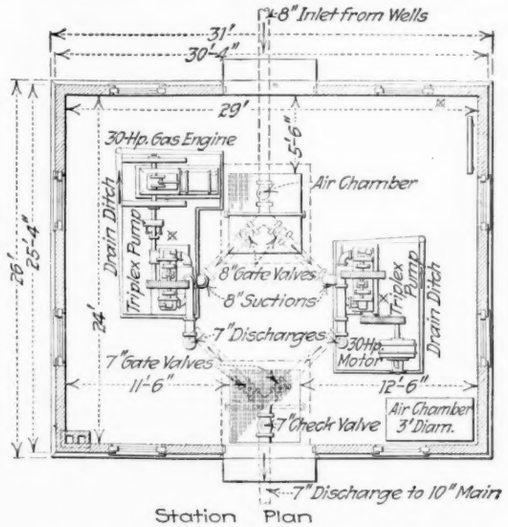

FIG. 3. PLAN OF THE PUMPING PLANT

start the pump at any time, although the tank may be completely filled. This eliminates the necessity for the use of a hand-operated means for starting the pump independently of the fall in pressure and results in the pump starting immediately in case of fire, thus materially increasing the reserve capacity available in the storage tank.

The contractor for the water-works was the Suburban Engineering Co., New York City; the automatic controller was built by the Automatic Switch Co., New York

Vulcan Soot Cleaner

The design of soot cleaner which has been gradually evolved by the Vulcan Soot Cleaner Co., DuBois, Penn., for Babcock & Wilcox type of boilers is shown in Figs. 1 and 2. The principle employed consists in directing steam jets from nozzles arranged along the length of cross-elements, down the diagonal path and at about 60 deg. to the horizontal. The element carrying these nozzles is rotated by a chain and sprocket wheel outside of the setting. In Fig. 2 the element has been rotated in the direction of the arrow through 90 deg. of its total travel of 180 deg. Part of the surface is shown as having been cleaned, and some of the tubes are to be cleaned when further rotation of the element will concentrate the steam blast upon them. Fig. 1 shows that there are two elements in each of the passes and also one for the superheater. These are all operated from the floor, the chains being hung to a convenient height and the main header and operating valve located about breast high.

The boiler is bricked up solid, the customary dusting-door panels having been eliminated. The cleaner is operated by first opening the main steam-supply valve in the steam line and blowing out any condensate. The main valve on the horizontal steam header of the cleaner is then opened, and the valves to each element operated in succession, commencing with that nearest the left end (Fig. 1).

Each sprocket wheel is rotated through its complete arc of 180 deg., this being done in a succession of small steps rather than in a continuous sweep.

Two other features are the air-relief valves and automatic telltale drain valve.

Early in the development of this cleaner it was found that internal corrosion of the outside valves and piping occurred, because when closing one of the operating valves the steam contained in the riser condensed, forming a partial vacuum, and as the piping was closed to the atmosphere, the pressure balance could be restored only by the piping being filled with the sulphur-laden furnace gases. The sulphur dioxide in these gases entered into combination with the moisture on the inner walls of the pipe, forming sulphuric acid, the presence of which caused serious erosion of the metal.

The trouble was corrected by the use of small automatic ball-check valves on all risers from the main header (Fig. 1). When the steam valve on any riser is closed, the ball check falls away from its seat, opening a minute orifice through which sufficient air is admitted to restore the balance of pressure within the riser and prevent the entrance of furnace gases.

The automatic telltale drain valve is similar in construction to the air-relief valve but larger, its purpose being to automatically discharge and indicate the presence of any small amount of steam or water that may leak from the main steam line and collect in the cleaner header. This valve automatically prevents the introduction of large slugs of water into the cleaner element and onto the tubes when the apparatus is next operated.

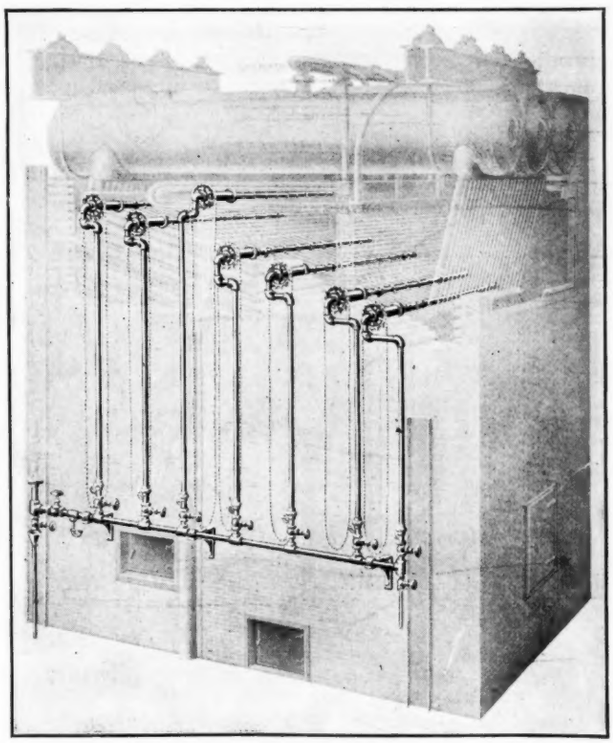

FIG. 1. WATER-TUBE BOILER WITH SOOT BLOWER

On standard Stirling boilers a cleaning system somewhat similar to that for Babcock & Wilcox boilers is used, with specially designed elements for the first pass, where high gas temperatures are encountered.

In order to clean the lower part of this bank of tubes a cleaner element is located so that the jets will have an effective range over the whole tube surface and at the same time be protected from the action of the heat of the furnace. This element is protected by the insulating wall which is built upon the bridge-wall and extends the full width of the setting.

The cleaning of the upper portion of the first bank of tubes presents a similar problem. This cleaner element has assembled on it return hay-rake bends terminating in two-way nozzles. Steam is turned into the element, which is then rotated bringing the nozzles up and over the baffle, behind which it has been protected, surrounded by stagnant gases. One outlet of each nozzle gives high steam velocity, as effective cleaning is desired at a distance; the other outlet is designed to spread the steam shortly after leaving the nozzle, to clean near-by surfaces. The nozzles are subjected to intense heat only while the steam is actually flowing through and cooling them. After the element has been operated, steam is shut off and the element is rotated in the reverse direction, so that the return bends and nozzles lie behind the protecting baffle.

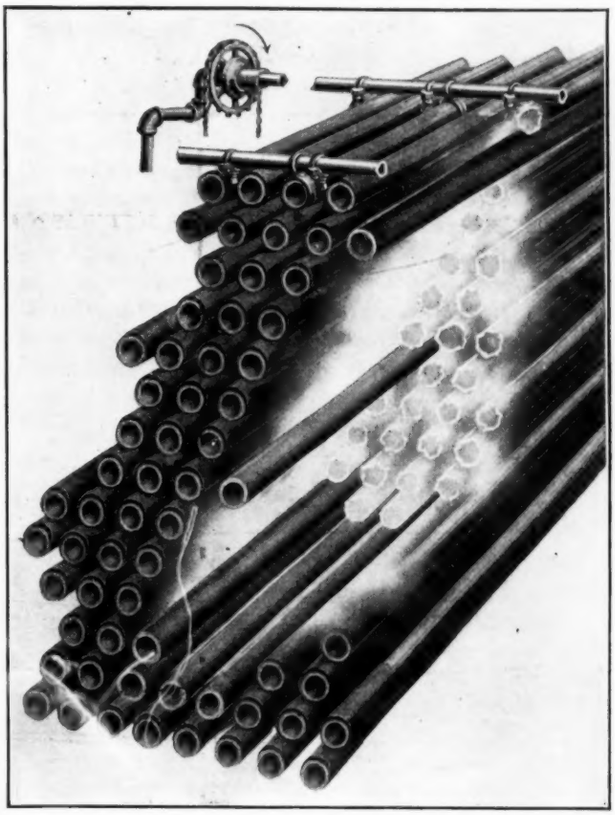

FIG. 2. SHOWING DIAGONAL PRINCIPLE OF CLEANING

This type of soot cleaner is also installed on economizers in several of the leading and latest power plants, and with the largest boiler units installed in the country.

Who Is Liable?—Can one man sell and another man buy and put into use a boiler that has been condemned as unfit for service, without incurring any civil or criminal responsibility? Baltimore has a boiler-inspection law that does not apply outside of the city. The city inspectors condemned a boiler in a plant in Baltimore, and the man put in a new one, selling the old one as a second-hand boiler, and it was installed in a factory on the eastern shore. About six months later the boiler blew up, killing six people outright, seriously injuring several others and causing considerable property damage.

A Study of Indicator Diagrams from a Compound Corliss

By Victor R. Gage*

SYNOPSIS—The diagrams, taken from the engine at comparatively light load, show that the logarithmic analysis cannot be applied under all conditions and also demonstrate the value of the temperature-entropy diagram as applied to such an analysis.

This test was made upon the high- and intermediate-pressure cylinders of a triple-expansion Corliss, thus making a 9x16x36-in. compound engine. The conditions of operation and the performance of the engine were: Steam pressure at throttle, 107 lb. gage and 43 deg. superheat; receiver pressure, 6 lb. gage; condenser, 23 in. vacuum with 29.26 in. barometer; 87 r.p.m.; indicated horsepower, 68.6; delivered horsepower, 64.3; mechanical efficiency, 93.7 per cent.; actual water rate, 16.36 lb. per i.hp.-hr.

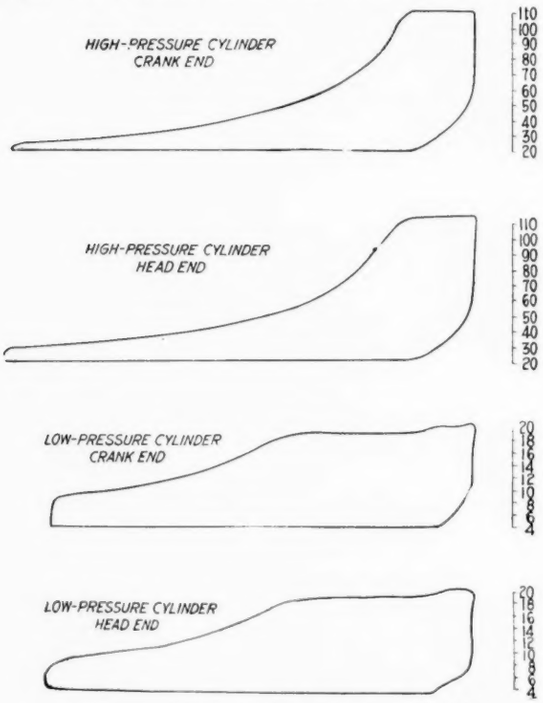

FIG. 1. ACTUAL INDICATOR DIAGRAMS

including jacket steam (or 13.7 lb. not including jacketing); distribution of work, high-pressure cylinder 48.5 per cent., low-pressure cylinder 51.5 per cent.; high-pressure cylinder not jacketed, receiver and low-pressure cylinder jacketed with steam from the main; condition of steam in the high-pressure exhaust, 95 per cent. dry, reheated in receiver to 40 deg. superheat; condition of low-pressure exhaust, 100 per cent. dry.

In order to obtain average indicator diagrams, the indicator pencil was held on the card for about two minutes, giving about 170 superimposed diagrams. The actual average indicator diagrams were carefully reproduced upon cross-section paper, using pressure and volume as coordinates, as in Fig. 2, and to logarithmic scale of pressure and volume in Fig. 3. Upon both of these PV diagrams the volume which would be occupied at various pressures by the weight of steam in the cylinder during expansion, provided it were at all times dry and saturated, is shown for both the head and the crank ends of the high-pressure cylinder (as the cutoffs were unequal on the high-pressure cylinder the weights during expansion are not the same) and for the low-pressure cylinder. The actual condition of the steam is shown by the quality curve on the high-pressure and by the superheat curve on the low-pressure cylinder.

On the logarithmic PV diagram[1], Fig. 3, the expansion lines are curved, not straight, lines. Hence the actual expansion of the steam in this case does not follow an equation of the form $PV^n = K$, where P is the absolute pressure, V the actual volume, n a constant exponent and K a constant. The curved form of the expansion lines might be attributed to leakage, or possibly to faulty application of the indicators. Neither of these explanations, however, seems logical in this case. The probable explanation is cylinder condensation and reëvaporation. The saturation curves are straight lines within the limits of the pressure range of the diagrams, so their equation is of the form $PV^n = K$. Now if an expansion line also had this same form of equation and n had the same numerical value, the two lines would be parallel or coincident, and the quality or condition of the steam would

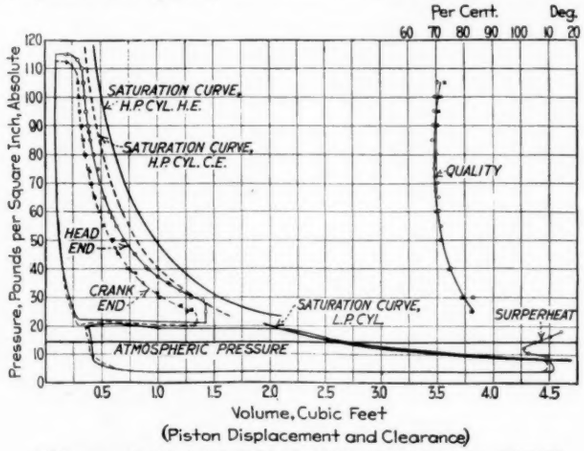

FIG. 2. AVERAGE DIAGRAMS, TOGETHER WITH SATURATION AND SUPERHEAT CURVES

remain unchanged during expansion. If the equations of the saturation and expansion lines were of this form and the exponent had different values for each, the two lines would be straight lines of different slope when plotted upon the logarithmic PV diagrams, and the condition of the steam during expansion would either continuously increase or continuously decrease in quality.

The steam begins to condense as soon as it enters the cylinder, continues to condense during all the period of

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[1]The reader is referred to "A New Analysis of the Cylinder Performance of Reciprocating Engines," by J. P. Clayton, University of Illinois Bulletin No. 58; also to "Power," June 18, 1912.

admission, and generally continues after the admission valve is closed until what may be called the mean temperature of the cylinder walls is reached; then reëvaporation begins. The point of thermal equilibrium depends upon several other factors besides the mean temperature of the surfaces. The temperature range of the steam, the size of the piston, the point of cutoff, the speed of the engine, all enter into the problem of the rate of heat transfer. If thermal equilibrium between the steam and the surfaces does not occur until some point on the expansion line is reached, it is not possible for the expansion line to plot as a straight line upon the logarithmic diagram.

This point of thermal equilibrium and the general behavior of the steam during expansion are shown readily if the temperature-entropy diagram is employed. Sometimes the temperature-entropy diagram is plotted from the entire indicator diagram. If this is done, it is necessary to assume that each cylinder is the boiler, the engine, the condenser and the feed pump combined; in other words, that the weight of steam in the cylinder remains constant at all times. On the temperature-entropy diagram, as plotted for this test, only the expansion lines appear, as the weight is constant during expansion. Also the condition of the steam supplied and exhausted from each cylinder is indicated; see Fig. 4. The steam at the high-pressure cylinder throttle is superheated 40 deg., but at cutoff the heat has been removed to such an extent that the steam is only about 70 per cent. dry. The initial condensation is a loss of great magnitude, and although nearly all of the heat is returned to the steam during part of the

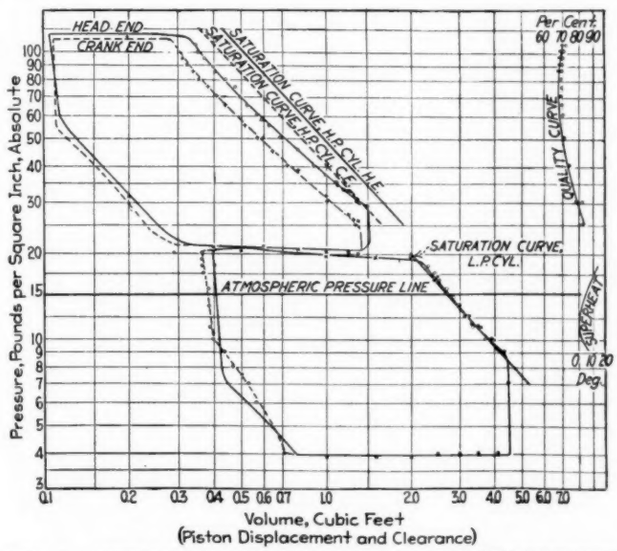


FIG. 3. AVERAGE DIAGRAMS PLOTTED TO LOGARITHMIC SCALE

expansion and all of the exhaust period, very little of it is available to do work in the cylinder.

That portion of the expansion line which is of constant entropy (vertical) indicates the period of thermal equilibrium during which the net interchange of heat between the steam and the surfaces is zero. From the high-pressure cutoff to the point of equilibrium the expansion line slopes downward and to the left, with entropy decreasing, indicating transfer of heat from the steam to the surfaces. Continuing below this point to release, the transfer is in the opposite direction. The quantity of heat transferred per pound of steam may be obtained by evaluating the area under the selected portion of the expansion line down to absolute zero (— 460 deg.). When the expansion line is vertical, the area is zero. The low-pressure cylinder expansion line shows constantly increasing entropy, indicating that at all times the transfer of heat is from the walls to the steam, as would be expected in a jacketed cylinder of this character.

There is a slight inherent error in computing the heat interchanges in this manner. The values of entropy are based upon the total energy required to bring one pound of water to the condition shown by a point on the diagram. When the cylinder is a closed vessel, the only heat energy

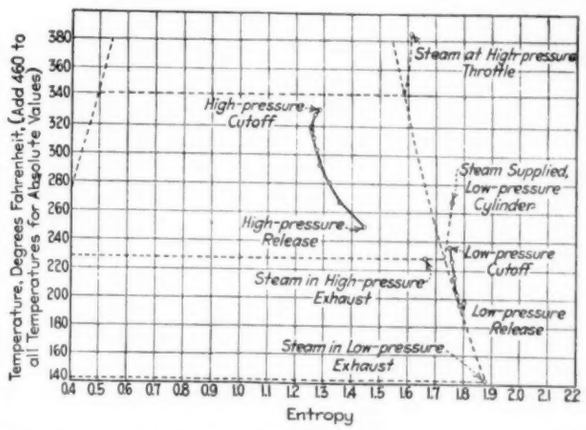


FIG. 4. TEMPERATURE-ENTROPY APPLIED TO DIAGRAMS

available to do work is the "intrinsic," or "internal," energy of the steam. Thus the heat interchanges as computed from the temperature-entropy diagram are based upon "total energy," whereas when computed by means of the more accurate Hirn's analysis, only the "intrinsic" energy is used. The numerical errors introduced in thus using the temperature-entropy areas probably are not of greater magnitude than the experimental errors. By means of Hirn's analysis the heat interchanges can be computed between admission and cutoff, between release and compression and between any points on the expansion line, or on the compression line. The advantage of the temperature-entropy method is in presenting, nearly correctly, the same results in a graphical form.

Specifications for Special Ammonia Pipe—Material to be uniformly good-quality soft weldable steel rolled from solid ingots. Sufficient crop shall be cut from the end to insure sound material, and the steel shall be given the most approved treatment in heating and rolling. All pipe 2 in. and larger to be lap-welded; smaller sizes to be butt-welded and redrawn from a larger size. Pipe must be reasonably straight and free from blisters, cracks and other injurious defects. Liquor marks incidental to the manufacture of lap-welded pipe will not be considered as surface defects. The pipe shall not vary more than 1 per cent. either way from being perfectly round or true to standard outside diameter, except on small sizes, where a variation of $^1/_{64}$ in. will be permitted. The pipe must not vary more than 5 per cent. either way from the weight specified. Where required pipe must have a good Briggs standard thread which will make a tight joint when tested by hydraulic pressure at the mill. The thread must not vary more than one and one-half turns either way when tested with a Pratt & Whitney Briggs standard gage. All burrs at the end are to be removed. Each length of National special ammonia pipe when lap-welded shall be tested at the mill to 2,000 lb. hydrostatic pressure; when butt-welded and redrawn, the test pressure shall be 1,500 lb. The steel from which the pipe is made must show the following physical properties: Tensile strength, 52,000 to 62,000 lb. per sq.in.; elastic limit, not less than 30,000 per sq.in.; elongation in 8 in., not less than 20 per cent.; reduction in area, not less than 50 per cent.—National Tube Co.

Rewinding Induction Motors

By G. Sabonjian

SYNOPSIS—On account of its simplicity the author suggests using a concentric distributed winding in place of a uniformly distributed winding for rewinding small size induction motors up to 5 hp. Also describes how to prepare the stator slots for the new winding and what kind of insulation to use.

Those who are not experts in the art of armature winding may find it difficult to place a uniformly distributed winding in the stator of an induction motor, especially in sizes below 5 hp. The following is a description of a method used in rewinding a 1-hp. two-phase machine that worked successfully. The original winding was wound with one layer of 24 coils uniformly distributed, 12 coils per phase, 3 coils per pole, as indicated in Fig. 1. The new winding was what is known as a distributed concentric winding, as in Fig. 4; this simplified the problem considerably as far as placing the coils was concerned. In placing the latter winding in the slots, all the coils of one phase are first put in their respective slots, and then the coils of the second phase. With the uniformly distributed winding, when the last coils are being placed, one side of the first five coils will have to be cut in order to get the last coils in under them, which makes quite difficult working.

Care should be taken in preparing the stator core for the new winding after the old one has been removed. The slots should be thoroughly cleaned of all old insulation or sharp corners on the laminations. When this has been thoroughly done, the next step is to insulate the slots. In the semi-inclosed type this can be conveniently done by making a fiber pocket that will just fit in the slot. This pocket is made by cutting a strip of $^1/_{64}$-in. fiber about ¾ in. longer than the slots and just wide enough so that when it is formed it will fit under the overhanging lips of the teeth. Fig. 2 shows the strip of fiber before it is formed. The pocket is shown in place at *A*, Fig. 3. Next a strip of oiled muslin is put in the slot, of such width that it will extend about ½ in. out on each side, as at *B* in the figure. This will assist in guiding the wires.

As will be seen in Fig. 4, the coils will have to be made three different sizes, but this is a simple matter as they are formed after they are placed in the slot, therefore, the only form that is necessary to wind them on consists of four heavy nails driven in a bench so as to give the coil the proper dimensions. The dimensions of the coil one way should be about equal to the distance between the slots that it is going to be placed in. For example, suppose the coil to be placed in slots *C* and *D*, Fig. 3, then its width should be made about equal to the distance between these two slots.

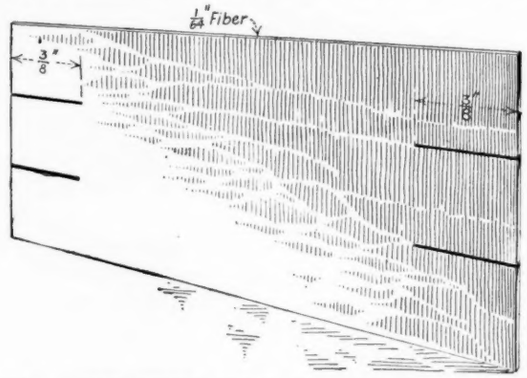


FIG. 2. THE FIBER STRIP FOR SLOT INSULATION BEFORE IT IS FORMED

The dimensions of the coil may be determined by measuring with a piece of wire which is placed around in the slots that the coil is going in and then cut the proper length; afterward it may be bent to about the shape of the coil and the form made accordingly. Care should be taken in determining the length of the coils, for if they are too short it will be difficult to get them into the slots, and if too long they will interfere with placing the bearing brackets.

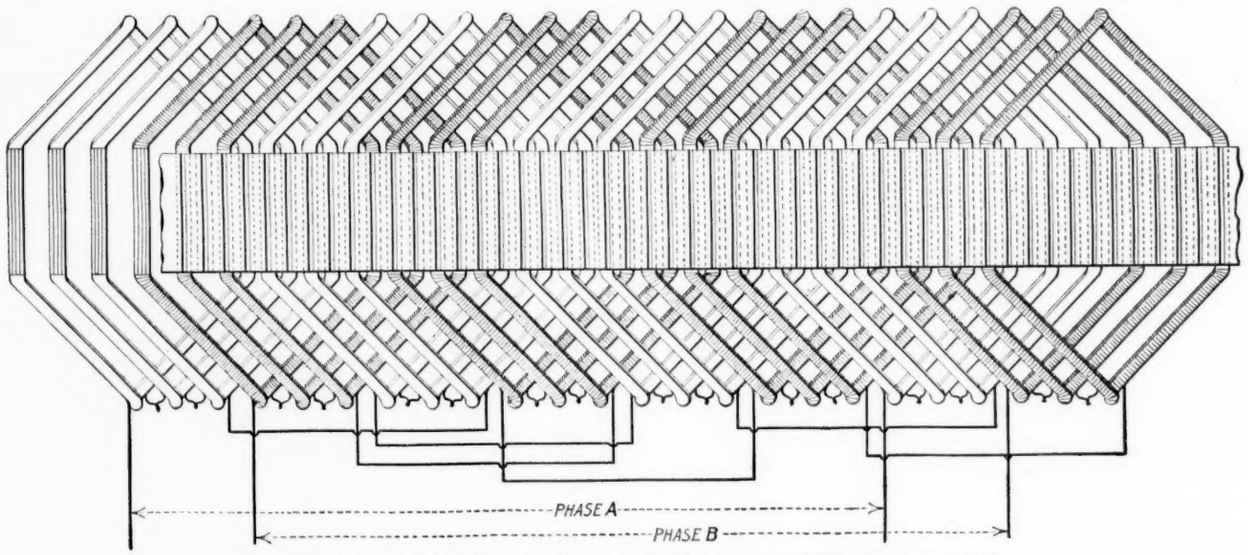


FIG. 1. FOUR-POLE, TWO-PHASE UNIFORMLY DISTRIBUTED WINDING

After the coils are wound, they should be tied at each end to prevent the turns from falling apart. This will give ample room to get the coils in the slots, which must be put in one wire at a time. When one coil has been placed, the part extending beyond the core is insulated with linen tape, and then the second coil placed and insulated in the same manner, etc. When the winding is completed, the muslin is cut off flush with the top of the slot and pushed down and a fiber strip driven in on top of it

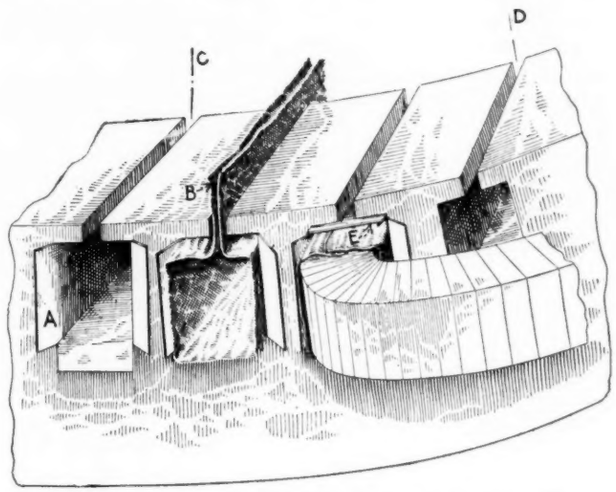


FIG. 3. SECTION OF STATOR CORE, SHOWING SLOT INSULATION AND COIL IN PLACE

under the slot lips, as shown at *E* in the figure. The first phase that is placed in the slots must be bent well down at the ends so as to allow room to get the second phase in the slots. When the winding is complete, the ends of the top coils should be tied down to the bottom ones and the whole winding given a coat of shellac or insulating paint and baked thoroughly in an oven for this purpose before putting the machine into service. In sizes above 5 hp. not much difficulty should be experienced in placing the uniformly distributed winding as the bore of the stator is of such diameter that will allow ample working space.

Design of Superheaters

By William C. Strott

I have been asked on several occasions what formula I use in calculating the amount of surface required for a superheater under varying conditions of service.

The subject is one that has not received the attention it should have by writers on the subject. Their articles have mostly so far only given an exposition of the various types, their construction, application, etc.

For the benefit of those who might be glad to add a good formula to their engineers' pocketbooks, I offer the following, with an example to simplify its application:

$$x = \frac{10S}{2(T - t) - S}$$

where

x = Square feet of superheating surface required per boiler horsepower;

S = Superheat required, deg. F.;

T = Temperature of furnace gas at superheater, deg. F.;

t = Temperature of saturated steam at boiler pressure, deg. F.

It is not often that the value of T can be readily obtained. For practical purposes it may be calculated from the following equation:

$$\frac{1}{(T - t)^{0.16}} = (0.172H + 0.294)$$

Where the values of T and t are the same as here given and H is the percentage of boiler-heating surface between the furnace and superheater, if there is 20 per cent. of the boiler-heating surface between the grates and superheater, then H will be eqaul to 0.20.

It is desired to find how many square feet of superheating surface is required for a certain boiler, carrying steam at 175 lb. (gage). Let H, as given, be 0.20. From a steam table it is found that the temperature (t of the formula) of steam at 175 lb. pressure is 378 deg. F. Substituting in the formula for T we have

$$\frac{1}{(T - 378)^{0.16}} = (0.172 \times 0.20) + 0.294, \text{ or}$$

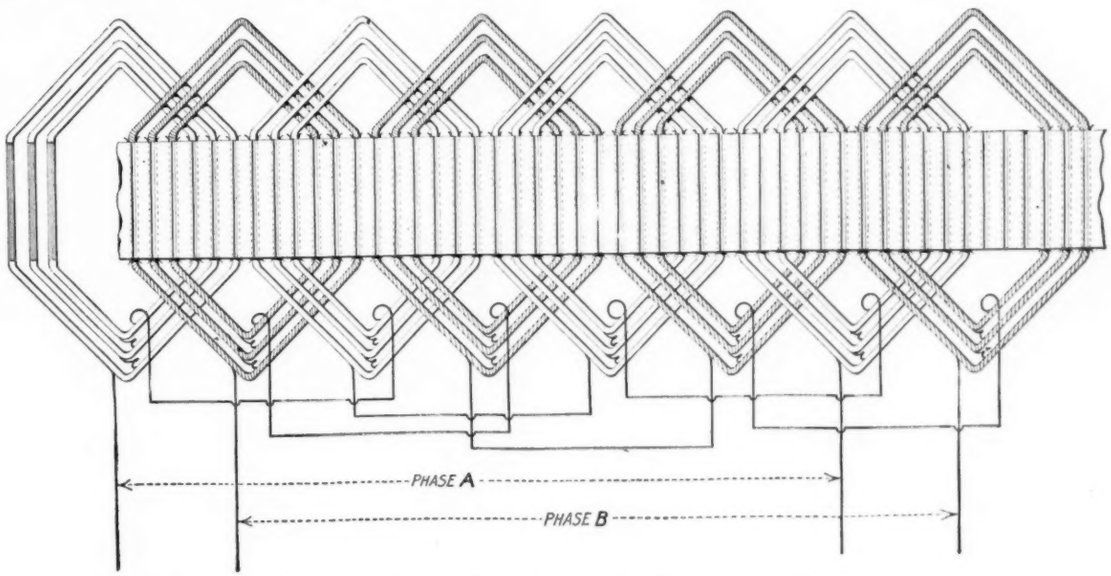


FIG. 4. FOUR-POLE, TWO-PHASE DISTRIBUTED CONCENTRIC WINDING

$$\frac{1}{(T-378)^{0.16}} = 0.3284, \text{ or } (T-378)^{0.16} = \frac{1}{0.3284}$$

$$= 3.084$$

$$(T-378) = \sqrt[0.16]{0.3284}$$

Now, $\sqrt[0.16]{0.3284}$ must be solved by logarithms, and is as follows:

$$(log\ 3.084) \div 0.16 = 0.489114 \div 0.16 = 3.056962$$

$$log\ 3.056962 = 1{,}140$$

therefore

$$T - 378 = 1{,}140$$

or transposing, we have

$$T = (1{,}140 + 378) = 1{,}518\ deg.\ F.$$

Suppose that we require 200 deg. of superheat which means that the temperature of the steam on leaving the superheater is to be 200 deg. higher than that corresponding to its pressure. We can now finish the problem.

$$x = \frac{10 \times 200}{2(1{,}518 - 378) - 200} = \frac{2{,}000}{2{,}080}, \text{ or approx. } 1$$

Therefore 1 sq.ft. of superheating surface is required per boiler horsepower. If our boiler is a 200-hp. unit, then we will require under the existing conditions, approximately 200 sq.ft. of superheating surface.

This formula assumes the steam as being a perfect gas. For every 1 per cent. of moisture in the saturated steam allow 15 per cent. more superheating surface than found by the use of this formula.

Troubles of a Refrigerating Troubleman

By D. L. Fagnan

After chasing Sam Bernard around the country for half a season, we were both in the same town on the same day and I had 'phoned and secured B-4 orchestra seat for that evening. The razor was keen and I gleefully took off the second coat when——. Well, 'twas a wire from the office. The Jones Cold Storage Co., Blankton, Tex., 800 mi. away, was in trouble. Leave at once. Fifty minutes later I was aboard the Limited and again Fate, steam and steel were increasing the distance between Sam and me.

The Jones plant was typical of those in Texas. A 50-ton compressor was driven by producer-gas engines. One 100-hp. 3-cylinder and an 80-hp. 2-cylinder engine were belted through friction clutches to the mainshafting. The layout was faulty in not having a friction clutch placed on the mainshaft to enable starting with a free shaft, gradually cutting in the compressor.

The trouble had to be quickly overcome, as some days it would be 9 or 10 a.m. before the plant could be made to run. Sometimes the engines would get a few good explosions and then pound. There would be heavy thumping in the compressor cylinder due to liquid ammonia. Then the operator would try to check the speed of the gas engine to minimize the thumping in the compressor.

I spent a day at the plant looking things over and laying out a plan of action. First we wired for the necessary fittings to make the bypass pipe full size. These arrived the following day by express and were installed that night, tested out and found all right. A friction clutch of the 6-arm type, known not to "lay down" on any job, was ordered. Of course it was ordered split. This was next installed, and as we had a fairly slack belt, we were able to easily run it off and on the pulleys.

Next the air cams on the engine were checked to see if full pressure of air was admitted at the proper time, namely, about 2 to 3 deg. early. We found them admitting 9 deg. early. The early admission when operating at such slow speed at starting neutralized a great deal of the energy of the air pressure. The camshaft was removed, and new offset keys were made and driven after careful measurements with a protractor had demonstrated that the angle was correct within ½ to 1 deg. The other engine was found to be taking air 12 deg. late, and as the exhaust valve opened at 44 deg. from the lower center, one can imagine what waste of air had occurred when I say that from a starting pressure of 160 lb. the pressure after one revolution was 100 lb. and after the second or third, about 50 lb., which was not sufficient to cause the engine to go over the top center. We reset the cams to take in air at 3 deg. early, which corrected the trouble had with starting.

Ignition System Needed Adjustment

Next the ignition systems were thoroughly gone over, one at a time. We found ignition taking place 18 deg. early at starting and when operating, 34 deg. early. I changed this to 7 deg. early for starting and 24 deg. for operating, with provision for easily varying the degrees by protractor readings on the shaft, leaving permanent marks; also on the eccentric cam I stamped degree marks.

The other engine ignited 10 deg. late at starting and about 12 deg. early when operating. This was changed to 5 deg. early at starting and 22 deg. early when running, which caused the engines to pull the load so much easier that we found we had ample power using either engine.

The next problem to solve was the reason why the operator could not get the proper temperature in the calcium-chloride brine tank. He could seldom get better than 0 deg. F. He informed us that he carried 15 lb. back pressure. There were two 25-ton vertical brine coolers. I determined to carry a low back pressure—about 2 lb.—and this gave a brine temperature of 15 deg. F. The frost was allowed to nearly cover the top pipe leading to the gages, showing the cooler contained liquid ammonia up to about two-thirds full.

This experiment over, I was told to overcome the chronic hot main bearing on the compressor. That night we dismantled the main bearing and took the weight of the shaft from it. Here was a serious case of flown babbitt. The oilways were stopped so badly that they were useless. The remedy here was obvious. Next we had to reline the friction-clutch bearings, which had heated considerably. As little care had been taken of the plant, the setscrews which prevented the lining shells from turning in the clutches, thus preventing the grease holes from matching, had come out and had never been replaced.

Manager Protests About Bill

I remained at the plant a little over two weeks and was busy every minute. Two nights I slept at the plant. But when I had finished, the plant ran well indeed and the manager was pleased. I was in town when the bill arrived, and he sent for me in a hurry. He wanted to know what the blankety blank, etc., dickens was the matter. He did not want any such bill. He would not pay it!

My company had charged $6 a day of 8 hr. and time and one-half for all overtime and Sunday work. He said he was paying his men $2.60 a day for 12 hr., and he had as good workmen as were to be found, etc. Of course he

would feel better if he exploded, so I patiently smoked during the disturbance. I determined to be real cutting—the situation demanded it—and asked why he sent for me and why some men get $10,000 a year while others get $750. After complaining about my hotel bill, meals, laundry, etc., he began to calm.

Then I asked him to figure out his previous losses under the old conditions, where his men stood around until 9 or 10 a.m. before they could start work. Now they started at 7. I reasoned with him and roughly figured the saving effected by my work. Shortly he began to feel bad for speaking so harshly. Well, to make a long story short, he took me motor riding and drove to his tailor, who measured me for a suit at my host's expense.

This plant is in Texas, and on my last trip through there I found things running fine and another engine of the same type as those installed.

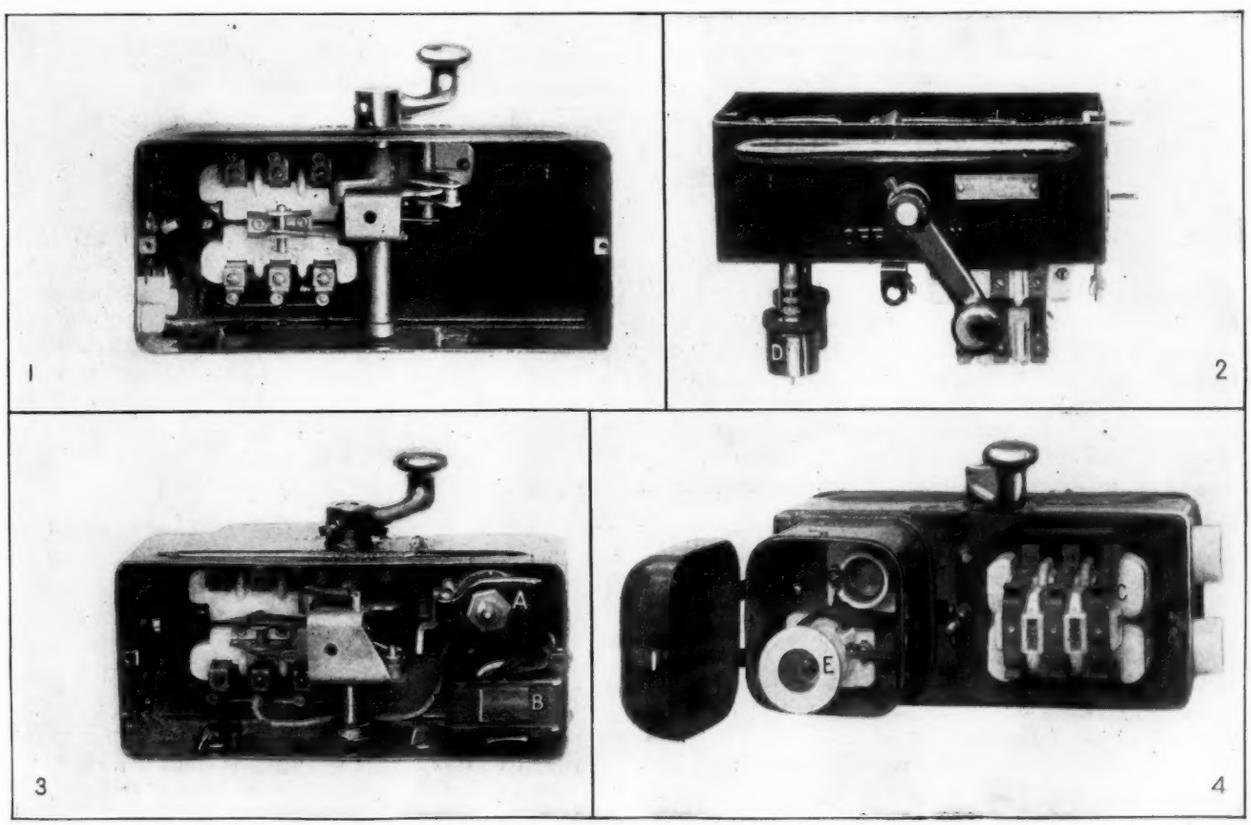
One may ask why they had to carry such low back pressures to get a low brine temperature. The engineer at the plant thought that the more liquid ammonia allowed to go to the brine coolers the lower the brine temperatures would be. He said he was told so by the erecting man who had installed the plant. It is true that 15 lb. back pressure corresponded to about 0 deg. F., but to get the proper flow of heat from the brine to the ammonia it was necessary to carry an ammonia temperature lower than zero, say from 5 to 6 deg., preferably 8 to 10 deg., so as to get the required temperature head.

Small Industrial Oil Switches

SYNOPSIS—A description of a new line of small industrial oil switches for starting and protecting three-phase induction motors up to 10 hp.

The switches here illustrated are used for starting and protecting three-phase induction motors up to 10 hp. and 600 volts; they conform strictly to safety-first principles. There are but few operating parts, and these are supported from the switch frame, which can be mounted on any flat vertical surface, such as a wall or post, or by supports on the motor-driven machines that the switches control. All operating or live parts, with the exception of the switch handle, are inclosed. The contacts are under oil; the terminals are in the interior of the frame, which is provided with a tight-fitting sheet-steel cover.

The switches are built nonautomatic (Fig. 1), with two-coil series-overload time-limit trip (Fig. 2), with plain low-voltage release (Fig. 3), and with combined low-voltage and overload protective plugs (Fig. 4). All switches are arranged for either open or conduit wiring. They open by gravity assisted by torsion springs and have a quick downward double-break per phase. The mechanism is accurately constructed and the parts are interchangeable. "On" and "off" on the frame indicate whether the switch is open or closed.



FIGS. 1 TO 4. DIFFERENT TYPES OF SMALL THREE-POLE OIL SWITCHES

Contacts, terminals, etc., are securely held in place by lock washers, preventing loosening from vibration. Stationary contacts consist of drop-forged fingers with flared ends. The movable contacts are copper strips bent to the form illustrated at *C*, in Fig. 4. This construction insures good contact under strong pressure when the

switch is closed. The oil tank is pressed from a single piece of sheet steel. A line indicates the depth to which the tank should be filled. The nonautomatic switch is held closed by a latch. In opening, the handle is moved some distance before the latch releases, after which the contacts are rapidly snapped open. The automatic switch, Fig. 2, with two-coil series-overload time-limit trip is opened automatically on overload by a trip-coil plunger. The switch cannot be held closed on overload on short-circuit. Overload trip setting can be varied between the normal current rating of the trip coil and twice normal by a calibrating device in the tube surrounding the lower

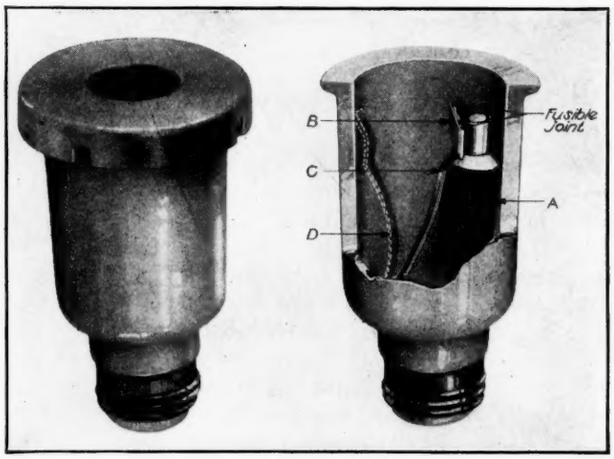


FIG. 5. TIME-LIMIT PROTECTIVE PLUG

portion of each trip-coil plunger. After a current setting is made, the device is locked in position by a setscrew. Each trip coil is individually calibrated, and the tube is then marked in legible white numerals on the black background of the tube.

For time delay in automatically opening the circuit, each overload trip coil is equipped with an inverse-time-limit attachment consisting of an oil dashpot, shown at *D* in the figure, connected to the tripping plunger. The time setting is accomplished without removing the dashpot from the calibrating tube. At proper setting, protection is afforded to the motor against overload and against running single-phase, but at the same time the switch is prevented from opening while the motor is starting. The calibrating tubes and oil dashpots are protected from injury by a sheet-steel case attached to the switch frame, which also prevents dust or lint collecting in the tubes.

The switch with low-voltage trip, Fig. 3, is tripped out instantly by a low-voltage release *A* if the line voltage falls to approximately 50 per cent. of normal. To reduce the watt loss of the low-voltage trip to a minimum, a small auto-transformer *B* is used in conjunction with the low-voltage release coil except at 110 volts 60 and 40 cycles, where the transformer is not necessary.

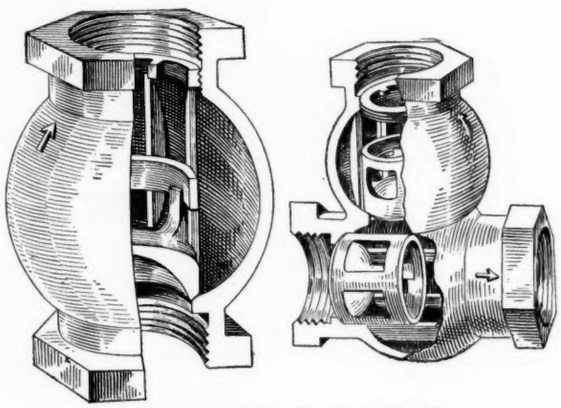
The switch with overload protective plugs and low-voltage trip, Fig. 4, is similar to the switches just described, but equipped with time-limit overload protective plugs *E* connected in series with the switch leads. In this case overload protection is obtained by two time-limit protective plugs as shown in Fig. 5. These protective plugs contain a stationary contact post with heating coil *A* and a fusible link *B*, which binds a spring contact *C* to the stationary post. The heating coil is in the motor circuit, which also passes through the post, link and spring contact. When the post is heated to a temperature that will melt the low-fusing alloy that holds the link together, the spring contact is released and takes the position shown by the dotted line *D*, thereby opening the circuit between the contactor and the post.

By reason of the time lag in the heating coil, the momentary inrush of starting current will not cause the plug to open the circuit. The plugs guard the motor against any conditions of overload that if maintained for a sufficiently long period would overheat the machine. Under the conditions obtaining when multiphase motors are running single-phase, or when an attempt is made to start stalled motors, or when a motor is called upon to drive too heavy a load, the plugs will open the circuit if properly applied. These switches are manufactured by the General Electric Co. and are known as their Type F, Form P-10 oil switch.

Smolensky Check Valves

The Smolensky check valve has been designed with only an inlet and an outlet, there being no cap through which the valve disk is inserted. The valve consists of three parts—a body, a disk and an adjusting ring to regulate the lift of the valve disk. On the interior of the valve body are ribs, which not only form a guide for the valve disk, but also give the liquid a whirling movement, causing the disk to revolve and seat differently. The seating face of the disk is rounded, but the valve seat, which is a part of the body, is a straight bevel.

Screwed into the threaded end of the valve body is the adjusting ring which is prevented from working back by the pipe end that screws into the valve end. The disk is made with an externally faced surface to slide on the ribs of the valve body and is thus guided squarely to its seat, no matter in what position the valve is placed. To the



TWO TYPES OF CHECK VALVE

top of the disk are projections that support a top guide ring. The spaces between the projections permit the flow of liquid through the valve. The general design is shown in the illustration. The valve at the right is a combined suction and check valve and does the work of two ordinary check valves. When a liquid is passing through the valve, the disk is balanced by the pressure surrounding it so that it does not bind in opening or closing.

The straight-way type is made in sizes from ⅛ to 24 in., as are also the suction and check valves. These valves are made by the M. Smolensky Manufacturing Co., Cleveland, Ohio, and handled by William Schliemann, 432 East 71st St., New York City.

Efficiency of Unsymmetrical Riveted Joints

By R. N. Blackburn*

SYNOPSIS—A discussion of the strength of riveted joints, particularly those not symmetrical in form. Exception is taken to the rule for such a joint, contained in the code prepared by the American Society of Mechanical Engineers.

In the Boiler Code issued by the American Society of Mechanical Engineers examples are given of riveted joints having apparent efficiencies of from 57.6 to 96.3 per cent. It is doubtful whether so high an efficiency as the latter figure is ever attained. It is practically impossible under workshop conditions to insure, for example, that each rivet shall fit its hole after driving with the same degree of tightness as every other, and it is evident that when a riveted joint is stressed, a tight-fitting rivet will take a greater proportion of the total load than one which is a looser fit. Obviously then the actual efficiency of the joint will be less than the calculated efficiency, which is based on the assumption that the stress is uniformly distributed.

But even when the workmanship is perfect, it will be found that the actual efficiency is considerably lower than the apparent efficiency as calculated by the formulas in general use. Such formulas are usually based on certain assumptions that are not strictly true, and the efficiency as thus calculated is therefore not the true efficiency of the joint. It is assumed, for example: First, that the total load is uniformly distributed over the rivets—that is, that each plane of shear of the rivets takes an equal proportion of the total load, or according to some rules, a definite proportion; second, that the stress in the shell plate along a line of rivet holes parallel to the joint is uniformly distributed over the net section of the plate, and third, that no bending stresses are produced in the plate by the pull on the joint due to the internal pressure in the boiler.

Assumptions Not Applicable

The first assumption is seldom true, the second one is never true, and the third can only be true for symmetrical joints, such as butt joints with straps of equal width and with all the rivets in double shear.

As an example of the unequal distribution of the load over the rivets, we will consider a double-riveted butt joint with butt straps of equal width, as illustrated in Fig. 1. The shell plate is ¼ in. thick, and as is usual with such thin plates, the butt straps are each of the same thickness as the shell plate. The total tension in a strip of plate of unit width—that is, of a width equal to the pitch of the rivets—may be assumed to be 3,000 lb. It is evident that the elongations produced by the load in those parts of the plate and butt straps between the inner and outer rows of rivets (denoted in Fig. 1 by *B*, *C* and *D*) are equal, and as the plates are of the same thickness, the total stress in them must also be equal. The tension on each must therefore be one-third of the total tension—that is, 1,000 lb. For the same reason the stresses in the outer and inner butt straps between the two inner rows of rivets, marked *E* and *F*, must also be equal, the tension in each butt strap being thus one-half of the total tension at *A*, or 1,500 lb. It follows, therefore, that a load of 1,000 lb. is transmitted by each of the rivets in the outer row to each butt strap, while a load of 1,500 — 1,000 lb., or 500 lb., is transmitted by each of the rivets in the inner row to each of the butt straps. The shearing stress in each of the outer rivets is therefore twice as great as that in the inner rivets. Therefore, if the nominal efficiency of the rivet section, based on the assumption that the load is equally distributed over the rivets, is, say, 70 per cent. of the strength of the solid plate, the actual efficiency is only 52.5 per cent.

Influence of the Strap's Thickness

In the example given, the difference between the nominal and the actual efficiency is due to the relative thicknesses of the butt straps and the plate. If we had assumed the thickness of the butt straps to be one-half that of the plate, the nominal efficiency and the actual efficiency would be the same. It is clear, therefore, that the relative thickness of the butt strap to that of the plate materially affects the efficiency of the joint, and it is to be noted that this factor is not usually taken into account in rules for calculating the efficiency of riveted joints.

Since the butt straps must be heavy enough for the joint to be properly calked, it is only practicable to proportion their thickness in accordance with theoretical considerations when the shell plates are very heavy. On account of other considerations it is usually advisable to make the thickness of the butt straps at least five-eighths that of the shell plates.

It will be found on consideration that this unequal distribution of the load over the rivets is more pronounced as the number of rows of rivets is increased, and for very wide joints, such as the quintuple-riveted butt joint illustrated in the Code of the American Society of Mechanical Engineers, one may safely conclude that the outer rows of rivets are carrying much more than their proper proportion of the total load. It is doubtful whether any appreciable increase of efficiency is gained by using more than three rows of rivets, and the high efficiencies claimed for such wide joints are largely imaginary. Moreover, the extremely wide inner butt strap required, which in the quintuple-riveted joint referred to would be nearly four feet wide, can scarcely be considered a desirable feature.

Stress Not Uniformly Distributed

Referring to the second condition usually assumed in formulas for the efficiency of riveted joints—namely, that the stress is uniformly distributed over the net section of the plate—a simple experiment will show that this is not true. If a narrow strip of elastic rubber be secured to a board by a nail near one end of the strip, a pencil line having previously been drawn across the rubber, as shown in Fig. 2, and if the strip be then

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pulled in the direction of the arrow, the pencil line will assume the position shown by the dotted lines, thus clearly showing that the stress is much greater near the nail than at the outer edges of the rubber strip, which, in fact, carry very little of the load. If the pull on the strip of rubber be increased so as to cause it to tear, it will be seen that the fracture always begins at the nail, proceeding outward in the direction of the lines A and B, a further proof that the stress is greater at the nail hole than at the outer edges of the strip. Owing to the greater flexibility of rubber as compared with steel, the stress appears more concentrated near the nail than it would be in a boiler plate, but there is no essential difference in the two cases.

Stress at Right Angles to Pull

The exact distribution of the stress in a boiler plate around a rivet hole cannot be directly determined by experiment, but there are good reasons for concluding that the distribution of the stress is such that the stress at any point in a line drawn at right angles to the direction of the pull in the plate through the center of the rivet hole varies inversely as the distance of the point from the center of the rivet.

To illustrate the effect of this unequal distribution of the load, suppose we have a flat steel brace, 3 in. wide by $\frac{3}{8}$ in. thick, secured by one rivet $\frac{15}{16}$ in. diameter, as shown in Fig. 3. The efficiency of the brace through the center of the rivet hole would, according to the usual rule for riveted joints, be taken as being equal to $\frac{p-d}{p}$; that is, $\frac{3-0.9375}{3} = 0.6875$.

If we assume the total pull on the brace to be 6,000 lb., the stress on the body of the brace would be equal to $\frac{6{,}000}{3 \times 0.375} = 5{,}333$ lb. per sq.in., and the net section of the brace through the center of the rivet hole is usually assumed to carry a uniform load of $\frac{6{,}000}{2\frac{1}{16} \times 0.375} = 7{,}757$ lb. per sq.in. As has been previously shown, however, the stress on the net section of the brace is by no means uniform, being greatest at the edge of the rivet hole and least at the outside edges of the brace. Referring to Fig. 3, the stress per square inch in the net section of the brace along the lines AB and CD varies from a maximum at the points B and C to a minimum at the points A and B, as shown by the small arrows; and on the hypothesis that the stress varies inversely as the distance from the center of the rivet hole, it can be shown that the stress at the points C and D is given by the following formula:

Stress per square inch at the sides of the rivet hole

$$= \frac{S}{td(hyp.\ log\ w - hyp.\ log\ d)}$$

where

S = The total load on the rivet;
d = Diameter of rivet;
w = Width of brace;
t = Thickness of brace.

It will be found on making the calculation that the stress per square inch at the points B and C is 14,652 lb., and as the stress per square inch varies inversely as the distance from the center of the rivet, it can be shown that the stress at the points A and B is 4,578 lb. per sq.in.

The actual efficiency of the net section of the brace along a line drawn through the center of the rivet holes at right angles to the direction of the pull is, therefore, $\frac{5{,}333}{14{,}652} = 0.3635$, which is only a little more than one-half the efficiency calculated by the rule generally used.

For a riveted seam in a boiler, the calculation of the maximum stress produced is complicated by the stresses caused by adjoining rivets. As comparative results sufficiently answer the purpose of this article, in order to simplify calculations as far as possible, in the subsequent consideration of the bending stresses in unsymmetrical joints, it will be assumed that the stress in the net section of the plate along the line of rivet holes is uniform. This assumption, which is made in all the rules in general use for determining the efficiency of riveted joints, will not materially affect the comparative results obtained.

To illustrate the magnitude of the bending stresses produced in an unsymmetrical joint, consider a unit width of the single-riveted lap joint illustrated in the

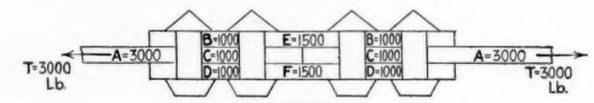


FIG.1

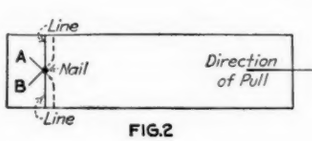


FIG.2

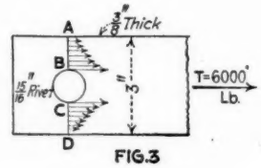


FIG.3

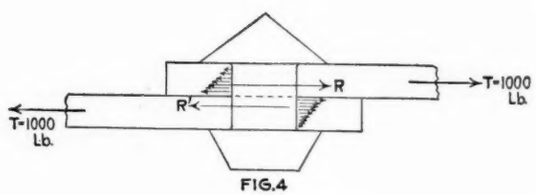


FIG.4

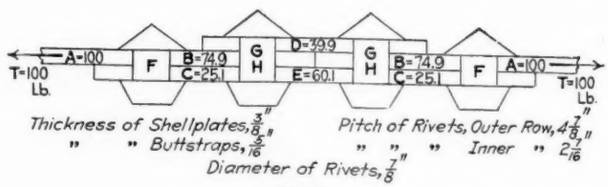


FIG.5

FIGS. 1 TO 5. APPLICATION OF STRESSES IN UNSYMMETRICAL BOILER JOINTS

Fig. 1—Unequal distribution of load on rivets. Fig. 2—Experiment with rubber strip and nail to show that stress is not uniformly distributed over net section of plate. Fig. 3—Effect of unequal distribution of stress. Fig. 4—Relative magnitude of pressure exerted by plates against rivets. Fig. 5—Numbers denote percentage of total load carried by respective parts of joint.

Boiler Code. Here the thickness of the plate is $\frac{1}{4}$ in., the pitch of rivets $1\frac{5}{8}$ in. and the diameter of the rivets after driving is $\frac{11}{16}$ in.

The efficiency of the net plate section through the line of rivet holes is given as 0.576. If we assume a pull of 1,000 lb. per unit width of joint, the stress per square inch on the net section of the plate along a line through the rivet holes, as usually calculated, equals 4,266 lb. per sq.in. This calculation does not take into account the stresses due to the bending action, which are more serious than the direct stress due to the pull on the plate.

In Fig. 4, which represents a sectional view of this joint, the pressures exerted by the plate against the rivet are indicated by short arrows. Owing to the tendency of the rivet to bend under the load, the pressure will be greatest at the plane of contact of the two plates and will continually diminish toward the outer surfaces of the plates. If we assume that the pressure continuously diminishes to zero at the outer surfaces of the plates, as shown by the length of the small arrows, then the resultants of these pressures, indicated by R and R', will act at points a distance of one-third the thickness of the plates from the surface of the joint. The bending moment thus produced is therefore 166.6 in.-lb., and the maximum stress produced in the plates by this bending action is 8,530 lb. per sq.in. If the pressures exerted by the rivet do not diminish to zero at the outer surface of the plate, the bending moment, and consequently the stress produced in the plate thereby, will be still greater.

The maximum total stress in each plate is therefore at least $8,530 + 4,266 = 12,796$ lb. per sq.in. This is about three times the calculated stress according to the usual method of calculation, and the true efficiency of the joint is, therefore, 19.2 per cent. instead of 57.6 per cent.

It may be mentioned here that in some actual tests of lap joints described in *Power*, Apr. 15, 1913, the maximum stress in the plate was found to be nearly four times that produced by the direct pull, which shows that the maximum stress as here calculated is well within the stress which may be produced in practice.

Other considerations affecting the strength of riveted joints are the bending stresses produced by the load on the rivets, and sometimes the rivets may also be subjected to a tensile stress.

From the foregoing it is abundantly clear that the nominal efficiency of a riveted joint bears no exact ratio to its real efficiency, and the fact that a certain type of joint may have a very high nominal efficiency does not necessarily prove that it is really more efficient than a different type having a lower nominal efficiency.

Lap Seam Properly Prohibited

That high stresses due to bending action are produced in lap-riveted joints is generally recognized, and the American Society of Mechanical Engineers has properly prohibited the use of this type of joint for large boilers. The only type of riveted joint in which no bending stresses are produced is a butt joint with inner and outer butt straps of equal width. It is to be regretted that the four examples of butt joints given in the Code recently issued by the American Society of Mechanical Engineers to illustrate the method of calculating the efficiency of riveted joints are all unsymmetrical joints in which severe bending stresses are produced. Although the unsymmetrical type of butt joint has a higher nominal efficiency than a butt joint with straps of equal width, the actual efficiency may be considerably less, but since the working pressure allowed on steam boilers is based on the nominal efficiency, it will naturally be concluded that the nominal efficiency of the joints illustrated represents a real and actual efficiency instead of being, as will be shown later, largely imaginary. It will be noticed, for example, that an efficiency of 80 per cent. is claimed for the double-riveted butt-strap joint illustrated in the Code. As the maximum efficiency possible for a double-riveted butt joint with butt straps of equal width and with all rivets in double shear is only about 75 per cent., we may inquire how far the efficiency claimed for the former joint represents the real efficiency.

The foregoing shows that the determination of the true efficiency of a riveted joint is by no means simple, but on the contrary involves some rather complex calculations. This is especially true where there are several rows of rivets and where, in addition to direct tension and shearing stress, bending stresses are produced. Then the calculations required are not only excessively laborious, but owing to the difficulty of ascertaining the actual distribution of the stresses in the joint, it is necessary to make certain assumptions and the results obtained by calculation are not, therefore, as satisfactory as the critical might desire.

It occurred to the writer that a fairly correct estimate of the stresses in a riveted joint of this type could be obtained by measuring the elongation under stress of the several parts of an india-rubber model of the joint. From the measurements thus taken, the stress in each part of the joint can be readily calculated and its efficiency determined. Several tests were made, and the mean results obtained are shown in Fig. 5. The numbers in the sketch denote the percentage of the total load carried by the respective parts of the joint. The shearing stresses on the rivets, expressed as a percentage of the total pull on a unit width of the joint, are as follows: Total shearing stress on each outer rivet at $F = 25.1$ per cent.; total shearing stress on each inner rivet at $G = 19.95$ per cent.; total shearing stress on each inner rivet at $H = 17.5$ per cent.

Proposed Method of Estimating Stress

There are two rivets in the inner row per unit width of joint, so that the total load on the inner rivets is double the amount given for the shearing stresses at G and H. As two of the three rivets in a unit width of joint are in double shear, the average load carried by each plane of shear equals 20 per cent. of the total load, and since the actual percentage of the total load carried by the outer row of rivets is 25.1, it is clear that the real efficiency of the rivet section is only $\frac{20 \times 100}{25.1}$ = less than 80 per cent. of the nominal efficiency.

If the total load per unit width of joint be represented by 100 units, the load per square inch in the solid plate at A is equal to $\frac{100}{4\frac{7}{8} \times 0.375}$ units, which equals 57.4 units, and the load per square inch on the net section of the plate along a line through the outer row of rivet holes, neglecting stresses produced by bending action, is $\frac{100}{4 \times 0.375} = 66.6$ units.

The stresses at A, B and C, however, produce a bending moment at F which causes a stress of 23.8 units per sq.in. in the shell plate at this point. The total stress in the shell plate at F is therefore $66.6 + 23.8 = 90.4$ units and the actual efficiency of the net section of plate at the outer row of rivet holes is therefore $\frac{57.4}{90.4} = 63\frac{1}{2}$ per cent. efficiency, as against a nominal efficiency of 82 per cent. Clearly, therefore, the above joint is inferior to a double-riveted butt joint with butt straps of equal width, which, since the latter joint does not cause bending

stresses in the plate, may have an actual efficiency as compared with the former joint of about 75 per cent. The unsymmetrical type of butt joint is inferior to the symmetrical type not only on account of its lower efficiency, but also because the breathing action produced by the bending stresses tends to cause a fracture in the shell plate along the outer row of rivets. In this respect the unsymmetrical butt joint is in the same class as the ordinary lap joint, which is now generally discredited for longitudinal seams of boilers.

A similar experiment made with a small-scale model of the quintuple-riveted joint illustrated in the Code showed an efficiency only slightly greater than that of the double-riveted butt joint just considered, the actual efficiency of the net section of plate at the outer row of rivets being only 66 per cent. instead of 96 per cent. as claimed for this joint. This low efficiency is due to the excessive proportion of the total load carried by the outer row of rivets and the correspondingly severe bending stresses produced at this point on account of these rivets being in single shear.

The fore part of this article showed that when there are several rows of rivets, the load is likely to be unevenly distributed, and on account of the extreme width of a quadruple-riveted joint the great difference between the nominal and the actual efficiency is not surprising. While the more accurate results that would be obtained by experiments with a full-sized model of this joint might possibly be somewhat more favorable, it is quite clear that the high efficiency of such a joint is largely imaginary.

Experiments Gave Approximate Results

Owing to the lack of suitable appliances, the experiments made were necessarily crude and the results obtained can be taken only as a rough approximation, but it is abundantly evident that unsymmetrical butt joints have a considerably lower efficiency than they are usually credited with and compare very unfavorably in this respect with butt joints having straps of equal width; and further, it is clear that little advantage is gained by increasing the number of rows of rivets beyond, say, three.

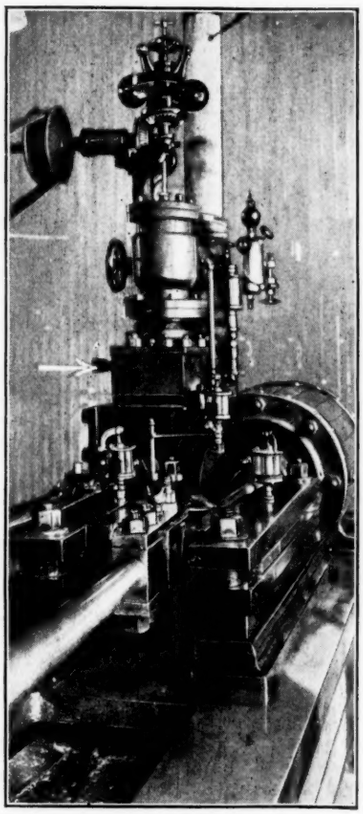
On account of the laborious calculations required, it is perhaps impracticable to attempt to calculate the true efficiency of riveted joints in everyday practice, and it is sufficiently near to the purpose to use the ordinary formulas, if a sufficient margin of strength is given by using an adequate factor of safety or, as it is sometimes justly termed, a "factor of ignorance." It is clear, however, that if the allowable working pressure is based on the nominal efficiency, a higher factor of safety should be used for unsymmetrical joints than is necessary for butt joints with straps of equal width. The Canadian Boiler Regulations, besides limiting the maximum nominal efficiency, provide for a higher factor of safety for lap joints and double-riveted unsymmetrical butt joints, and although the increase in the factor of safety required is not sufficiently great to cover the excessive bending stresses in this class of riveted joints, the adoption of symmetrical joints is at least to that extent encouraged and the inferior unsymmetrical type officially discountenanced.

Unless some distinction is made between these two classes of joint, manufacturers will naturally adopt the type having the higher nominal efficiency in spite of the fact that such efficiency is largely imaginary, and the unsymmetrical butt joint will thus become the standard used, in preference to the more efficient and decidedly superior symmetrical butt joint. At present not only have boilers with unsymmetrical joints a lower real factor of safety than is perhaps generally recognized, but a relatively excessive value is placed upon wide joints and the boiler with the superior symmetrical butt joint is placed at a commercial disadvantage.

Another Old Engine

By R. E. Keech

The engine illustrated herewith was built by Woodbury, Booth & Co., of Syracuse, N. Y. (now out of business). It originally had a cast-iron crankshaft, but later on a steel shaft and disk were substituted. The 10-ft. diameter cast-iron flywheel has a rim 4 in. square. The engine is now run at from 145 to 150 r.p.m., driving a 45-kw. d.-c. generator for the lighting station at Saranac, Mich., a town of about 900 population. The engine is of the riding cutoff-valve type. A lever on the steam chest, shown by the arrow, operates a valve to throw the steam behind the first valve or the second one, according to the load on the engine. There are a large number of old engines doing daily duty throughout the country that are in better shape than some of the more modern designs that have been in operation but a few years, and are a credit to their builders.



OLD-TIME ENGINE

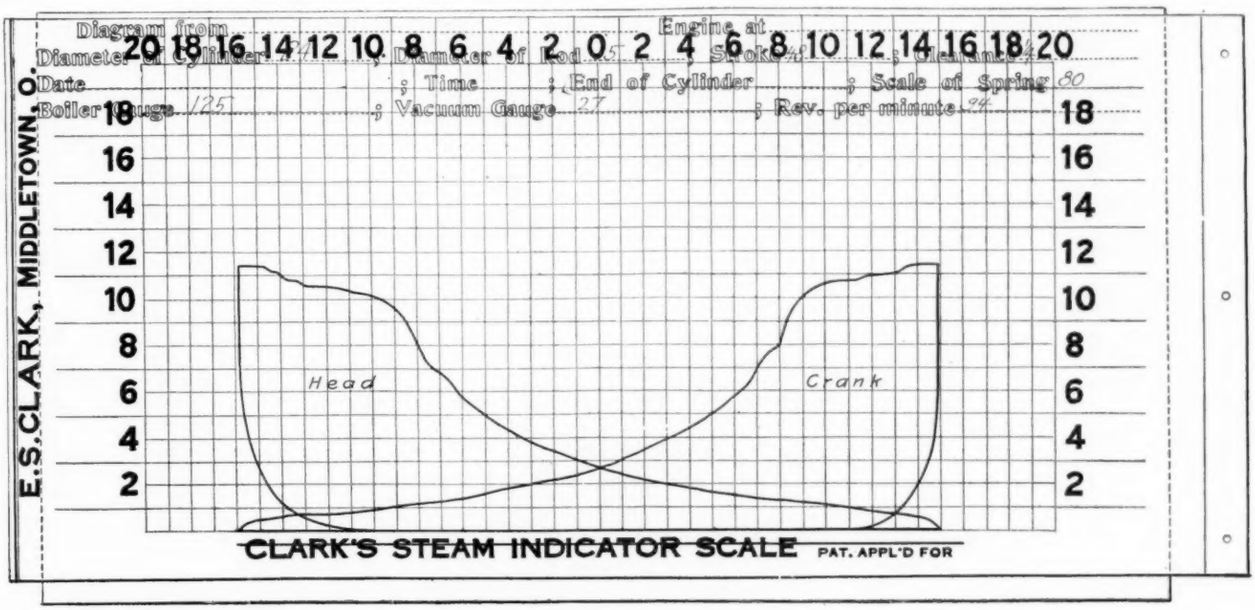
Steam Indicator Scale

A useful device has been got up by E. S. Clark, Middletown, Ohio, to aid engineers in computing their indicator diagrams. By its use one can determine the accuracy of the valve setting, the equality of clearance and obtain assistance in figuring the mean effective pressure of indicator diagrams. It is also useful in making a comparison of the expansion curve with the hyperbolic curve.

The illustration shows an indicator card that is placed between a metal frame and a transparent cover, lined and figured as indicated. The vertical lines are used for graduating the card and are numbered each way from the center, or zero, line. The horizontal lines

indicate the pressure at different points and are numbered from the bottom, or zero, line upward.

When used for valve setting and equalizing the clearance, the card is placed with the back-pressure line under the horizontal zero line, with the center of the diagram at the vertical zero line, as shown. A comparison of the figures at the beginning and ending of each curve on one end of the card with those at the other end



INDICATOR DIAGRAM APPLIED TO THE SCALE

will show how closely they correspond. If they are equal, the valve setting is also equal. If the clearance is equal, the compression curve will correspond with the figures at each end of the card; if not, the clearance is unequal and the compression curve will not correspond.

When used to find the m.e.p., the card is placed as shown, following the steam line and the expansion curve, using every other vertical line. For instance, the crank-end steam line shows 14.8 on the vertical line, the admission line shows 11.5, and the compression ceases at 6. The line 14.8 is therefore $11.5 - 6 = 5.5$ lb. Again take the vertical line 12 that shows a pressure of 11, the back pressure 0.1, then $11 - 0.1 = 10.9$ lb. The same idea is carried out for the remainder of the diagram. It will be noted that the value of the vertical lines is in pounds pressure instead of in inches as in measuring the ordinary ordinates. The sum of the pressures as indicated by the various ordinates for each diagram divided by the number of ordinates is the mean effective pressure.

A booklet accompanies the scale, giving directions for its use.

Computation of the Flow of Steam or Other Gas in pipes involves the density (lb. per cu.ft.) or specific volume (cu.ft. per lb.) taken as the mean for the pipe length (not the density at the initial or the terminal pressure). This may or may not coincide with the middle section of a given run of pipe, for if there are numerous fittings at one end and few in the other, the pressure drop and consequently the density will vary accordingly. To be exact, the temperature as well as pressure must be considered, for if the temperature is higher than that due to the pressure—superheated by external means or by the friction of flow—its density will be less.

A Close Shave for "Whiskers"

By Jack L. Ball

"Speaking of luck and safety valves," said the Chief, "brings to my mind an incident that happened about ten years ago out there in that boiler room.

"I was on night shift working twelve hours, and on this particular night we had been up against all kinds of trouble—hot boxes and the like—and I'll admit I was not in the best of humor. Just as I sat down to eat my lunch, one of the seediest-looking and most-bewhiskered individuals that I have ever seen walked in and asked if he could stay over night. I disposed of him in short order, but after he had gone, I felt just a little ashamed of myself as it was raining furiously.

"The night wore on, and one of the firemen reported No. 5 feed-water regulator working badly. On this boiler we had a few days before put a new 3-in. spring-loaded safety valve and had not yet connected up the escape pipe, and when it popped the steam blew up against the brick wall of the boiler room five or six feet from the valve. Going over on top of the boiler, the first thing I saw was my long-whiskered visitor of the early evening, lying fast asleep right in front of the outlet of the safety valve.

"Well, when I found that fellow asleep there and courting death, I was not lenient with the toe of my shoe. Just as I came down after adjusting the regulator so that the stem would not bind and stock, the cooking vats were cut off. The fires were heavy and No. 5 safety valve, being a little light, let loose with a roar at 150 lb. and a temperature of over 350 deg. F. I have often wondered if that fellow ever knew just how close he came to being cooked."

Automobile Registration—Statistics compiled by the United States Department of Agriculture show that there were 2,372,696 automobiles registered in the United States in 1915, netting a total revenue of over eighteen million dollars in registration and license fees. This record is directly traceable to the high stage of development of the gasoline engine.

Boiler Tests with Fuel Oil

The activity in the burning of oil fuel under boilers will likely make unusually interesting the accompanying table of tests of oil burning Babcock & Wilcox boilers at the Navy Yard, Philadelphia, Penn. The table is from a paper on "Oil Fuel," by Ernest H. Peabody, presented at the International Engineering Congress, San Francisco, Calif., September, 1915.

FUEL-OIL TESTING PLANT, UNITED STATES NAVY YARD, PHILADELPHIA, PENN.

1	Test No.	93	94	95	96	97	98	99
2	Date, 1913	July 16	July 16	July 17	July 17	July 23	July 24	July 25
3	Barometer	30.12	30.035	30.01	29.96	30.085	29.855	30.035
4	Length of test averaged, hr.	2	3	3	3	3	4	3
5	No. of burners used	3	5	8	10	8	10	13
6	Aggregate opening of air registers, sq.in.	77.1	128.5	205.6	270	205.6	270	334
	Steam Observations:							
7	Steam pressure, lb. gage	300	300	300	300	300	300	302
8	Calorimeter, lower therm. deg. F.	310	309	286.2	267.3	303.1	292.3	272.7
9	Calorimeter, normal reading, deg. F.	312	312	312	312	309	309	309
10	Feed temperature at boiler, deg. F.	199	200.6	201.3	200.6	200.2	200.7	200.6
11	Water actually evaporated, lb. per hr.	13,052	23,644	37,504	48,216	36,638	47,662	60,325
	Oil Observations:							
12	Oil burned, lb. per hr.	871	1,578	2,532	3,192	2,486	3,182	4,005
13	Oil pressure at burners, lb. gage	191	218	196	207	188	194	193
14	Oil temper at burners, deg. F.	180	180	180	180	180	180	181
	Air Observations:							
15	Dry bulb, deg. F.	78.9	84.1	78.8	79.8	87.2	86.0	80.35
16	Wet bulb, deg. F.	65.3	67.5	67.6	68.4	72	73	64.5
17	Fireroom temper., deg. F.	97	99.7	94.3	96.6	101	98.4	96.9
18	Stack temper., deg. F.	446	509	620	685	649	690	720
19	Fireroom pressure, in. of water	1.00	2.00	3.26	4.25	3.21	4.20	5.11
20	Velocity head, by pitot tube, in. of water	0.29	0.73	1.66	2.48	1.67	2.54	3.516
21	Smoke, Ringelmann scale	0—2	0—1	0—1	0—1	0—1	0—1	0—1
	Flue-Gas Analysis, from First Pass:							
22	Carbon dioxide, per cent.	12.625	14.266	14.2	14.66	14.34	15.106	15.15
23	Oxygen, per cent.	5.04	2.71	2.75	2.15	4.34	3.093	2.73
24	Carbon monoxide, per cent.	0.00	0.00	0.00	0.00	0.00	0.00	0.00
25	Nitrogen (by difference) per cent.	82.335	83.024	83.05	83.19	81.32	81.801	82.12
	Calculated Results:							
26	Steam quality	0.999	0.9982	0.985	0.974	0.9966	0.9903	0.9789
27	Factor of evaporation	1.0684	1.0662	1.0545	1.046	1.0652	1.0594	1.0501
28	Actual evaporation per pound of oil, lb.	14.984	14.983	14.812	15.104	14.738	14.98	15.062
29	Equivalent evaporation per pound of oil, lb.	16.01	15.98	15.62	15.80	15.70	15.87	15.82
30	Actual evaporation per sq.ft. of h. s.	3.263	5.911	9.376	12.054	9.159	11.915	15.081
31	Equiv. evaporation per sq.ft. of h. s.	3.49	6.30	9.89	12.61	9.76	12.62	15.84
	Oil Results:							
32	Oil burned per sq.ft. of h. s., lb. per hr.	0.218	0.3945	0.633	0.798	0.6215	0.795	1.001
33	Oil burned per cu.ft. of furnace, lb. per hr.	1.95	3.54	5.69	7.18	5.58	7.16	9.00
34	Oil burned per burner, lb. per hr.	290	315.6	316.5	319.2	311	318.2	308
	Air Results:							
35	Relative humidity, per cent.	47.5	42.0	55.5	55.3	47.7	53.2	42
36	Dew point, deg. F.	57.2	58.5	61.5	62.5	64.6	67.0	54.7
37	Moisture, lb. per cu.ft.	0.0007495	0.000788	0.0008705	0.00090	0.0009646	0.000555	0.0006928
38	Weight of air at atmospheric conditions, lb. per cu.ft.	0.07375	0.0729	0.07385	0.07375	0.07265	0.0725	0.074
39	Forced draft blower output, cu.ft. per min.	6,280	9,910	14,700	18,000	14,860	18,340	21,380
40	Air supplied to burners, cu.ft. per min.	3,360	5,710	9,125	11,450	9,335	11,840	13,980
41	Velocity through vanes of air registers, ft. per min.	6,280	6,400	6,400	6,110	6,540	6,320	6,030
42	Cu.ft. of air per pound of oil.	231.2	217	216	215	225.2	223.2	209.6
43	Cu.ft. of air theoretically required for complete combustion of one lb. of fuel	193.3	196.5	194	194.2	197.2	197.5	193.5
44	Excess air, per cent.	19.7	10.5	11.3	10.7	14.25	13.0	8.3
	Results from Dry Gases and Ultimate Analysis:							
45	Lb. of dry gases per pound of fuel.	17.156	15.263	14.645	14.878	15.23	14.49	14.44
46	Lb. of dry air per pound of fuel.	16.98	15.15	15.23	14.78	14.77	14.10	14.11
47	Per cent. excess air over amount theoretically required for complete combustion.	30.66	14.3	14.53	10.95	25.72	13.92	14.62

HEAT BALANCE

Test No.		93 B.T.U.	93 Per Cent.	94 B.T.U.	94 Per Cent.	95 B.T.U.	95 Per Cent.	96 B.T.U.	96 Per Cent.	97 B.T.U.	97 Per Cent.	98 B.T.U.	98 Per Cent.	99 B.T.U.	99 Per Cent.
48	Heat utilized	15,536	79.57	15,507	79.42	15,157	77.64	15,331	78.52	15,235	78.04	15,400	78.88	15,351	78.64
49	Loss by imperfect combustion	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
50	Loss with dry chimney gases	1,472	7.54	1,535	7.15	1,857	9.51	2,152	11.02	2,052	10.51	2,111	10.81	2,210	11.32
51-a	Loss with moisture from hydrogen in the fuel	1,366	.....	1,396	.....	1,463	.....	1,495	.....	1,471	.....	1,496	.....	1,514	.....
b	Loss with moisture in the fuel	6	7.13	6	7.28	6	7.64	6	7.80	6	7.69	6	7.77	6	7.87
c	Loss with moisture in the air	20	.....	20	.....	23	.....	23	.....	25	.....	14	.....	17	.....
52	Radiation and error	1,125	5.76	1,061	6.15	1,019	5.21	518	2.66	736	3.76	498	2.54	427	2.17
53	Total	19,525	100.00	19,525	100.00	19,525	100 00	19,525	100.00	19,525	100.00	19,525	100.00	19,525	100.00

Reducing Glare of Light Bulbs

If prussian blue be applied to the light bulb as far down as necessary to prevent the direct glare of the light from shining in the eyes, it will be found much more comfortable for the workman. As is pointed out by Fred Fruhner, this will throw a bluish light to the eyes, but the direct rays of the light are thrown on the work. The blue should be evenly laid on.

To Preserve Timber from Decay it is treated with an antiseptic, such as creosote or zinc chloride. Creosote is the oldest known preservative and one of the best. Creosote oil is insoluble in water, and has a boiling point of over 4,100 deg. F.

Direct-Current Motors of the Lower Horsepowers

To meet the increasing demand for direct-current motors of the smaller horsepowers for direct connection and drive of all kinds of industrial machinery, machine tools and the many motor applications requiring a low-power compact motor, the C & C Electric and Manufacturing Co., Garwood, N. J., has developed a new line.

The new motors, known as the Type IB, are of bi-polar type, with interpoles, and can be furnished in ratings up to 10 hp. in either the shunt or compound wound. For a general manufacturer's drives, as machine tools and other industrial machinery, the characteristics of the shunt-wound motor are the most desirable.

Coal Production of Iowa in 1915, according to figures just made public by the United States Geological Survey and the Geological Survey of Iowa, amounted to 7,614,143 short tons, valued at $13,577,608, which represents an increase of 163,121 tons, or 2 per cent. in quantity and $213,538, or 1.6 per cent. in value and was exceeded in only two previous years, 1909 and 1910. The number of men employed in the coal mines of the state decreased from 16,057 in 1914 to 15,549 in 1915, but the number of days worked per man increased from 204 to 220.

Editorials

Educational Work in Engineers' Organizations

In the organizations composed chiefly of power-plant engineers of the operating class, little is done in an educational way in the warm summer months. It is during this lull that instructors and educational committees should plan the work for the coming season, for if this is left until the last minute, lack of a good start will retard the season's progress.

In such organizations the local sections, branches and associations plan and carry out most of the educational work carried on by them. Those acquainted with the situation reluctantly confess that most of those put in charge of such work do not take their jobs seriously enough. Perhaps the greatest mistake is made in not laying out and having available at the beginning some progressive course of study that may be carried on during the season so that by the time the season ends the members will feel that the winter's work has been worth something. When members feel this way, it encourages greater attendance at the meetings, and the enthusiasm and goodwill of the satisfied members are two of the greatest factors in getting increased membership. For the body to meet with no subjects set aside for discussion, but to rely upon someone to "bring up" something, is like a man aimlessly walking the streets with no particular destination. Considered from the standpoint of efficiency in management, such methods cannot be too strongly condemned.

Many members are greatly dissatisfied because the business transacted at the meetings, which in themselves could be dispatched quickly, are frequently drawn out because of inexcusable harangue over questions of parliamentary procedure. Every organization that suffers from this evil should call together strong and influential members and find some way of eliminating it. If this harangue were to stop at parliamentary discussion and go no further, it would not be so bad. The truth is that jealousy is encouraged, and it is this jealousy that is the stab in Caesar's back and the open wound that bleeds the organization to its lowest vitality.

All good things cost something. The instructor or committee cannot lay out a comprehensive and beneficial course of study without some money to defray expenses. There should be no hesitancy about appropriating as large a fund for educational work as can possibly be spared. If the members could feel at the beginning of the season or at the time they are asked for such an appropriation (or it may be an assessment out of the ordinary), reasonably sure that the results were going to be worth the investment, it would be found that nearly all would willingly give. The instructor or the committee, however, must convince the members that results can be had.

Surely, the subjects from which to select lectures, discussions and blackboard talks are numerous enough in the power-plant field to give variety to the season's work. A canvass among the members will likely reveal that an all too small percentage make use of the free public libraries. This condition is deplorable, especially in the large cities where the library authorities have established technical sections in the libraries and have well-arranged and tabulated and invariably well-selected books relative to the trades.

One of the things most discouraging to an instructor or to a committee is the reluctance on the part of men who can do so, to rise and participate in the discussion of subjects on which they are well informed. This characteristic does not indicate that these men have information which they will not give away, but it is the manifestation of what one might term "nerve inertia." Experience teaches that about the only way of overcoming this difficulty is to repeatedly call on these men and get them started to participate in discussions and thus slowly wear away their reluctance.

Those responsible for educational campaigns will often find that it expedites the work to set aside a number of subjects and, after carefully considering the qualifications of the different members, assign members to subjects that they are most competent to discuss. Usually, when these men are given to understand that they are expected to develop the subjects given and when they hear of their assignment for the first time as it is announced to the organization in meeting, they will accept.

When one considers the position of the operating engineer in the sphere of social economy and surveys the competitive forces developing on all sides of him, one wonders that so little time is given to education and so much to business in other directions the productive value of which is often negative.

In these organizations there are many men who must be forced to a realization of their need of greater education. Showing these men that this is so is a disagreeable job, and it requires on the part of the instructor or the committee a high sense of duty to incur the ill-will of these men by telling them the truth about themselves in language plain and unveneered, for " 'tis sometimes necessary to be cruel to do good." But this injured pride is passing and soon turns into deep appreciation.

Boiler Explosions Here and Abroad

American engineers have been led to believe, and doubtless with valid reasons, that steam-boiler operation in England is carried on under a much more efficient system of inspection and supervision than is the general practice in this country.

In comparing the number of boiler explosions that occurred during the year 1915 in England and in the United States, one should take into consideration the difference in area of the two countries and the number of boilers in operation. A notable fact in regard to boiler explosions in this country in recent years is that they are largely confined to sawmill and similar plants in which skilled engineers are not employed and in which

explosions the larger number of persons are killed and injured.

Disastrous boiler explosions, such as those that occurred at St. Louis, Brockton and Chicago several years ago, have been conspicuous by their absence during the past few years, which would imply that the lessons presented as a result of these disasters have been heeded.

According to the annual report of the Committee of Management of the Manchester Steam Users' Association, there were noted throughout Great Britain during the year 1915, forty-eight boiler explosions, killing twenty-two persons and injuring fifty-four others. The association, which by the way has had but one fatality during its sixty-one years of operation, had under inspection over ten thousand boilers. In Massachusetts during the same year over twenty-six thousand boilers were inspected. By this it will be seen that the State of Massachusetts alone has about two and one-half times as many boilers under inspection as are inspected by the Steam Users' Association.

During the past twelve months there were in the United States, in so far as we were able to learn, two hundred and sixty-five boiler explosions or failures of all kinds, in which one hundred and thirteen persons lost their lives and three hundred and twelve others were injured. About one-fifth of these explosions were confined to low-pressure house-heating boilers; thirty-three were sawmill boilers and these killed twenty-eight and injured fifty-five others.

The Secretary of the Board of Trade, Manchester, reporting upon the working of the Boiler Explosions Act, shows that there were sixty-one explosions during the year ending June 30, 1915, resulting in the loss of twenty-two lives and injuries to fifty-five persons. It would appear from these figures, as well as from those of the Steam Users' Association, that there are as many boiler explosions in Great Britain as in the United States in proportion to the number of boilers in use. Deducting the number of house-heating boilers from the list of explosions for the year 1915 in this country, the total of high-pressure explosions is reduced to about two hundred and ten. This is about three and one-half times the number of explosions that occurred in Great Britain during the same period.

From the foregoing it would appear that as favorable a showing has been made in this country as has been made abroad. In fact, the figures credited to Great Britain seem to be excessive, and one is led to wonder if the war, which has doubtless called many engineers to the front, is not to some extent responsible in that less experienced men have been employed in their places.

⁂

Protecting Electric Machines

In the pioneer days of the electrical industry the problem of control and protection was a simple one. A switch to open and close the circuit and a fuse to automatically disconnect the generator in case of trouble were all that was necessary. Continuity of service was of somewhat secondary importance compared with the protection of the generator, consequently when a short-circuit occurred on the system, it was a case of getting it off as quickly as possible even if this required disconnecting the generator.

In the modern electric-power system, instead of one low-voltage generator of small capacity, several high-voltage alternators of very large capacity are operated in parallel in a single station. With a number of these stations tied together in one great network of over half a million horsepower, the problem of protection and control becomes of an alarming magnitude. Some of the modern systems are capable of instantaneously delivering millions of horsepower into a defect which, if not properly protected, would cause serious destruction.

Great damage might be done if the protective devices failed to work in case of trouble on such vast concentration of energy, supplied from modern alternators that will stand enormous overloads for short periods on even a dead short-circuit without injury. That in many cases the greatest damage was done to the system in clearing the trouble made it evident that the need of protection was not so much for the generator as for the other parts of the system. This, along with a demand for uninterrupted service to the customer, has brought about a radical change in the control and protection of electric equipment.

This change has been a complete reversal from the old idea of getting the circuits open as soon as possible in case of trouble to keeping closed during the abnormal conditions all switches except those that are necessary to disconnect some part of the system that has been permanently disabled and limiting the energy that may flow into the defect by current-limiting reactances. In short, the modern idea is to segregate the trouble to the part of the system where it belongs. This is accomplished by the use of an elaborate arrangement of relays operating oil switches in the proper sequence. The outcome is that the protection is given to the various parts of the external circuit and not to the generators except in case of trouble within the machine itself, when in many cases the generator switches are nonautomatic from any external trouble.

⁂

Now that high combustion chambers, more correct air supply and distribution, mechanical stokers and other means for improving combustion are more and more widely applied, one wonders if tubes as large as four inches in diameter are necessary in boilers of the usual length, and if smaller tubes would not improve the transmission of heat. The gas itself is a poor conductor, and as efficiency is so greatly dependent upon the closeness, the rapidity and the area of contact of the hot gas and the tube, a tube that will be large enough to care for draft conditions and yet small enough to insure that a minimum of gas gets through without making contact is the most desirable one. Not forgetting the differences in volumes of gas and water to be cared for, condenser experience, where, neglecting air leaks, high vacuum for a given amount of water circulated depends almost solely upon heat transmission, has shown that a great number of small tubes providing a maximum of surface for contact give best results. One should not neglect to observe, however, that the boiler must have room in and between the tubes for adequate circulation of both water and gas.

⁂

Remember that in legal disputes over matters of boiler construction the Boiler Code is likely to be more and more regarded as the most authoritative authority.

⁂

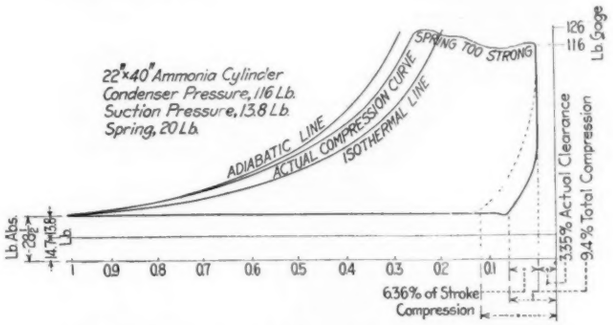
Many steam engineers never really appreciate what the loss of a seemingly few B.t.u.'s per hour or per day mean until they calculate the refrigerating capacity of circulated brine.

Correspondence

Displacement of Ammonia Compressors

Reading Thomas G. Thurston's letter on "Displacement of Ammonia Compressors" in the Mar. 21 issue of *Power*, page 413, it strikes me that this subject is worthy of more discussion than it has received in issues which have since appeared. Clearance in ammonia compressors seems to become more general, not only for reasons of safety, but also for efficiency.

In analyzing Mr. Thurston's article, I am unable to agree with his illustration of the loss of horsepower through clearance. If the gas left in the clearance space did not lose any heat by reëxpanding, then the figures given by Mr. Thurston would be correct, but, as he is aware, possibly 10 per cent. of the energy stored in the gas is lost in transmission, and in giving up the energy consumed during compression, it is accompanied by loss of heat. If this is the case, then the increase of volume caused by reëxpansion cannot be as large as illustrated



SHOWING CLEARANCE AND COMPRESSION

by Mr. Thurston; that is, in the case of 185 lb. condenser pressure and 15.67 lb. suction pressure, the increase in volume cannot be 6.66 times. Therefore I am of the opinion that just as much as the pressure drops sooner by reëxpanding, the suction valve must open just so much sooner, and the effective displacement of the compressor will be just so much greater and the horsepower lost through clearance will be just so much less.

The accompanying reproduction of an indicator diagram from a 22x40-in. double-acting dry ammonia gas compressor with 3.35 per cent. clearance illustrates what effect a large clearance has on the general efficiency of a compressor.

This machine has two double-acting water-jacketed dry gas compressors, and is nominally rated at 475 tons at 56 r.p.m. Operating conditions at the time this diagram was taken were as follows: Condenser pressure 131 lb. abs., suction pressure 28.5 lb. abs., 45 r.p.m. The dotted line represents the reëxpansion curve if no heat had been lost. This is the same curve Mr. Thurston used in his illustration.

To make comparison, calculated the same way as Mr. Thurston did: Swept volume = 15,205 cu.in.; 3.35 per cent. clearance = 510 cu.in.; total volume = 15,205 + 510 = 15,715 cu.in.; ratio of reëxpansion = 131 ÷ 28.5 = 4.6 times increase of volume; 510 × 4.6 = 2,346 cu.in. of reëxpanded gas in clearance space.

For two compressors and 45 r.p.m. the theoretical amount compressed and reëxpanded per 24 hr. would be (2,346 × 4 × 45 × 60 × 24) ÷ 1,728 = 351,900 cu.ft. The amount of ammonia required per ton per hour at 131 lb. abs. pressure and 28.5 lb. suction pressure is 25 lb. The volume of 1 lb. of gas at 28.5 lb. abs. pressure is 9.55 cu.ft. The volume to be handled in 24 hr. at this pressure to produce a ton of refrigeration will be 24 × 25 × 9.55 = 5,730 cu.ft.

The horsepower per ton for this machine at the pressures given is approximately 1; in other words, the horsepower required to compress 5,730 cu.ft. of gas in 24 hp.-hr. To compress 351,900 cu.ft. will require (1 × 24 × 351,960) ÷ 5,730 = 1,474 hp.-hr. Assuming 19 per cent. loss of mechanical energy, then 19 per cent. of 1,474 is 280 hp.-hr. Of the total volume of 15,715 cu.in., 2,346 cu.in. is reëxpanded gas from the clearance space, leaving a net volume of 15,715 — 2,346 = 13,369 cu.in. of new gas in the cylinder per stroke. The total volume handled per 24 hr. will be (4 × 45 × 60 × 24 × 13,369) ÷ 1,728 = 2,005,350 cu.ft. The power consumption will be (24 × 2,005,350) ÷ 5,730 = 8,400 hp.-hr. The loss from 3.35 per cent. of clearance will be 280 ÷ 8,400 = 3.33 per cent. of the total input. With steam consumption of 20 lb. per hp.-hr. at a cost of 20c. per 1,000 lb. and assuming a run of 250 days per year, the loss will amount to (280 × 20 × 0.20 × 250) ÷ 1,000 = $280, if figured according to Mr. Thurston's method.

The results obtained as calculated from the indicator diagram are as follows: The gas in the clearance space reëxpanded 2.9 times equals 510 × 2.9 = 1,477 cu.in.; the amount of gas compressed and reëxpanded per 24 hr. for the two compressors at 45 r.p.m. is (1,477 × 4 × 45 × 60 × 24) ÷ 1,728 = 221,383 cu.ft.; to compress 221,383 cu.ft. of gas requires (1 × 24 × 221,383 ÷ 5,730 = 927.26 hp.-hr. Nineteen per cent. for loss of mechanical energy = 0.19 × 927.26 = 158.18 hp.-hr.

Of the total volume of 15,715 cu.in., 1,477 cu.in. is reëxpanded gas from the clearance space, leaving a net volume of 15,715 — 1,477 = 14,238 cu.in. of new gas in the cylinder per stroke. The total volume handled per 24 hr. will be (4 × 45 × 60 × 24 × 14,238) ÷ 1,728 = 2,135,700 cu.ft. The power consumption will be (24 × 2,135,700) ÷ 5,730 = 8,945.34 hp.-hr. The loss from 3.35 per cent. clearance will be 158.18 ÷ 8,945.34 = 1.77 per cent. of the total input.

With steam consumption of 20 lb. per hp.-hr. at a cost of 20c. per 1,000 lb. and assuming a run of 250 days per year, the loss will amount to (158.18 × 20 × 0.20 × 250) ÷ 1,000 = $158.18, as figured from the indicator diagram.

Now $280 — $158.18 = $121.82 less than figured the theoretical way and shows conclusively that the losses

attributed to clearance in ammonia compressors are not as large as generally believed.

My opinion is that an ammonia compressor with a reasonable amount of clearance and constructed to insure safety from knocking out cylinder heads and the falling of broken valves into the cylinder handles a greater weight of gas than, although not as large a volume as, a nonclearance compressor, and this to some extent, counterbalances the loss of fuel.

New York City. WILLIAM SCHLIEMANN.

Navy Boiler Compound Used at Fruitvale

On page 894 of the issue of June 20, the second part of a report of the Station Operating Committee of the National District Heating Association refers to the use of Navy boiler compound at the Fruitvale power station of the Southern Pacific Co., but the discussion seems to be based on a misconception of the method of treatment, as well as operating conditions.

The water fed into the boilers at Fruitvale is not maintained at an alkalinity of 3 per cent. normal, but the water in the boilers is maintained at 3 per cent. normal. It means, therefore, that the water in the boiler carries 92 grains per gallon excess soda ash and that it does not concentrate beyond that point. The density of the boiler water seldom runs higher than 1.0016. Since the treatment was begun (March, 1914), there has never been a tube failure due to density, and the few tubes now renewed from time to time are chiefly on account of replacing handhole covers or junction boxes. The construction of the boilers is such that in order to remove either it is necessary to cut out and lose the corresponding tubes.

Reference is made to the chance at Fruitvale of seam-lap cracks, tears, ruptures, etc., due to the use of the Navy compound or to the method of treatment. The following shows the analyses of three of the leading boiler compounds sold on the Pacific Coast, as compared with the Navy:

Compounds	1 Per Cent.	2 Per Cent.	3 Per Cent.	Navy Per Cent.
Sodium carbonate or hydrate	43.34	43.65	37.95	76.00
Trisodium phosphate	8.47	24.40	22.28	10.00
Soluble organic	7.66	8.64	21.13	3.00
Water and impurities	40.53	13.31	18.64	11.00
	100.00	100.00	100.00	100.00

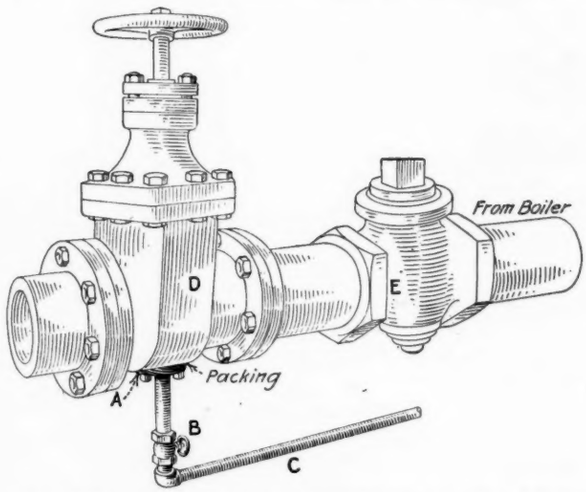
Tests on the boiler waters of plants using these compounds, as well as at plants using only soda ash (the basis of them all), have shown alkalinities in many cases as high as those carried at Fruitvale, and in some cases much higher. The treatment has been in force at Fruitvale for only a little over two years, but the conditions at the other plants have existed for many years and without any signs of the alleged troubles.

It is not believed that Mr. Babcock's paper referred to can be taken as meaning that no other cure could be found for the trouble at Fruitvale, but that at that date no other cure had been found. It is now the general opinion of those familiar with the conditions, that any of the compounds tried during the early days at Fruitvale would have corrected the trouble if used under the present method of treatment. The Navy compound is now being successfully used in some forty-odd boiler plants and ferryboats, under as many varying water and operating conditions. FRED E. GEIBEL.

San Francisco, Calif.

Clearing a Blowoff Valve

The illustration shows a gate valve on a blowoff line in the bottom of which a small quantity of sediment remained every time it was used, preventing it from closing tight. The remedy, as told to me by a friend, was to drill a 3/8-in. hole in the bottom of the valve at the center, bolt a small flange *A* to the bottom of



LIVE STEAM TO CLEAR BLOWOFF VALVE OF SCALE

the valve as shown, and attach a 1/4-in. steam pipe *C* with a valve *B*. The plug valve *E* was first opened, then the live-steam valve *B*, and lastly the gate valve *D*. In closing, the order was reversed—first *D*, then *B*, and lastly *E*.

Live steam blowing through keeps the large valve free from sediment at the bottom pocket. Whether the method is the best that can be applied, I do not know.

Toronto, Ont., Canada. JAMES E. NOBLE.

Underloading Oil Engines

In the June 20 issue R. E. Mathot mentions underloading as being detrimental to the proper working of Diesel or hot-bulb engines.

By reason of several years' experience in operating an engine of the latter class (De la Vergne type F. H.), I feel sure that Mr. Mathot is overestimating the probabilities of trouble from this source. The engine of which I have charge is 90 hp. and has been in operation five years. During each year of the five, at least a month's run has been made with a load not exceeding 20 hp., this light load period being in advance of the heavy operating season. There has been no accumulation of carbon or soot on the piston head or in the clearance spaces, and the internal surfaces are so clean today that only a faint smudge is left when the hand is rubbed over them.

I believe that the best indication of complete combustion in any Diesel or semi-Diesel engine is the exhaust; and if this be true, then the theory of incomplete combustion at light loads is exploded, for during such periods much less trouble is experienced in keeping the exhaust clear than during normal or extremely heavy loads.

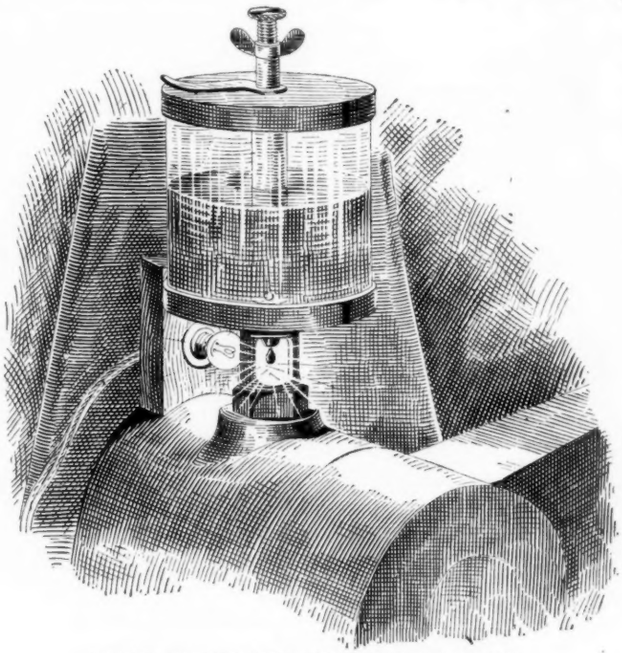
Our fuel has ranged from 18 deg. to 28 deg. Baumé California asphalt-base oil and has never cost much above 2c. per gal. Contrary to the general opinion that such engines soon lose in economical operation, we are still

able, after five years' wear on valves, cylinder and piston rings, to come well under the manufacturer's guarantee of ½ lb. per brake horsepower-hour. From these facts we are led to believe that any marked loss in economy on this type of engine indicates poor mechanical condition due either to overloading or neglect by those in charge.

Pomona, Calif. AMON J. FORTNER.

Illuminating a Sight Feed

Lubricators are frequently placed in dark places, and it is hard to see the drop of oil in the sight glass. This difficulty may be overcome by placing back of the glass a miniature electric lamp. If but one light is used, a



MINIATURE ELECTRIC LAMP AT OILER

battery is best for furnishing the current, but if several oilers are closely grouped, the lights can best be connected in series and hooked up to a regular circuit through a switch so that the current can be turned on or off at will. T. W. BUERSTATTE.

Pullman, Wash.

Stop Wastes of All Kinds

One is apt not to realize that vats or tubs for boiling processes will waste more steam than they should, when supplied from a live steam line. In a certain factory where live steam is used in open vats to keep a liquid boiling for several hours, the power department complained that when these vats were on, it was difficult to keep up steam because it was being wasted. A steam-flow meter on the line to one of the vats showed that with the steam valve open wide, as had been the custom, 60 boiler horsepower was consumed. By trial it was then found that by throttling the supply the liquid could still be kept boiling with 45 b.hp. and the desired results obtained.

We now have a meter on the main supplying all vats, the flow is regulated according to the number of vats in service and the factory is charged with the steam by this meter. Care is therefore taken not to use more than is necessary.

This is an example of steam wasted in factory operation. Steam costs real money, and when a saving is made it proves that the engineer who is wide awake is worth more to his company than the one who lets the waste continue. Take a look around your own factory and see if you can stop a waste. H. A. HOFFMANN.

Philadelphia, Penn.

Rating and Guarantee of Small Centrifugal Machinery

In the Apr. 4 issue there appeared an editorial under the heading, "Unfulfilled Guarantees." This has naturally caused considerable comment, particularly among manufacturers on account of the unpleasant truths it contained. From a broader standpoint, however, the article is a particularly timely one, and the publication of a few more like it would do more than anything else to eliminate unfair and unwholesome competition.

During the last few years I have repeatedly argued against the practice of giving too serious consideration to the maker's paper guarantees. The editorial hits the nail right on the head when it says, "and a reminder to the purchaser that the writing of a guarantee does not establish its performance." Again, instances were quoted where the paper guarantees were missed by 20 and 25 per cent. Surely this does not reflect on the manufacturer only, but on the whole engineering profession. If the purchasing agent insists on buying on the basis of paper guarantees only and the salesman has reason to believe that the machine will never be tested, the temptation to put in an absurd figure to beat the other fellow becomes so strong that he will often do so and take a chance.

Looking at the situation from a manufacturer's standpoint, competition is increasing daily and the cost of labor and material is going up without a proportionate increase in the selling price. The size of the machine must therefore be cut down so that it will just fill the requirements and no more. The large overload capacities that were common with small turbines a few years ago are no longer commercially practicable. There is a movement on foot to sell small turbines on a maximum rated basis, similar to the now standard practice in larger machines and electric motors. This subject will be referred to later.

Returning to the question of guarantees as applied to auxiliary apparatus and small installations, of how much real importance are they to the customer? Of course one must discriminate between a machine of average efficiency and a steam hog, and our arguments will be confined to machines of the former class. Where all the exhaust steam can be used, a big variation in steam consumption is permissible without affecting the coal pile to any appreciable extent. Again, how often do small auxiliaries work under the conditions for which they were designed? In the majority of cases a small part of the time, either the overload valves are open when running at half load or the steam and back-pressure conditions are entirely different from those originally specified.

Apart from the relation, if any, between the guaranteed and actual performance, the practice of making impossible guarantees should be more discredited than it is by most people at the present time.

With all due respect to the designers of small turbines, pumps or blowers, the guarantees for efficiency made by them must necessarily be considered in the light of the best guess possible based on interpolation or extrapolation. The turbine maker has by far the best argument in this respect, however, as he has done much more than anyone else to determine the true performance of his apparatus. He usually has an elaborate testing equipment, and practically all guarantees can be verified before shipment if necessary. So much cannot be said, however, for the pump and blower maker.

The designer often resents any suggestion that he is taking chances when making guarantees that are just a little better than we have been accustomed to, but if he would only look at the situation as others see it, based on common horse-sense reasoning, and state frankly that "this is what we expect to get," and show sound reasoning why they should get this figure, there would be more respect paid to the estimated guarantees than is paid to the absolute guarantees at the present time.

If the designers of the two ends of a piece of apparatus were to take each other more into their confidence and each show a disposition to help the other fellow, more cordial relations would exist between the turbine maker and the manufacturers of pumps and blowers. As the situation now stands, the pump maker figures an efficiency of 67 per cent. This requires, say, exactly 21 hp. The price has been cut to the last cent, and the turbine supplied will deliver the 21 hp., and that's all. The machine falls down a few per cent. in capacity after installation, resulting in months of controversy and several harsh letters between the respective builders before the matter is adjusted. This condition of affairs has cost them in addition thousands of dollars, which could have been saved by a broader conception of what is known of the performance of centrifugal machinery.

All Guarantees Are Uncertain

To go a step farther, if the respective makers would get together and educate the customer to the fact that there is an element of uncertainty in all guarantees, although the designers are making the best possible use of the data at their disposal, there would be a tendency to give more leeway in the acceptance of the machine. As it is, one of the customer's junior engineers, a narrow-gage man, will probably run the test, and because the machine falls down two or three per cent., he will recommend rejection, as the guarantee was absolute and he is fulfilling his duty to his employer. Even if the broader-gage man higher up sees it in the same light as the manufacturer later on, considerable money has probably been spent in time and expenses by officials of the company in accomplishing this end. There is no one to blame but the manufacturer himself, however, for giving in the first place as absolute a guarantee.

By taking the broader stand, no reflection on the ability of the respective designer should be warranted. On the contrary, he should receive the greatest possible credit for coming as near as he has done to his guarantees in the past, with the limited data at his disposal. The layman naturally cannot see it in this light. He feels that because the manufacturer has been in the business for so many years, he should know by this time exactly what to expect. Even the executive heads of our manufacturing plants fail to properly realize the amount of work entailed and the time necessary for the engineering department to produce for the sales department data that will cover intelligently and accurately all the conditions that they are called upon to cover by a guarantee.

It has already been stated that the turbine manufacturer is better posted than the builders of any other type of centrifugal apparatus. Let us consider what he is up against to formulate a line of salesmen's data. Suppose, for the sake of argument, it is decided to run an exhaustive series of tests on one size of wheel under conditions called for by present commercial requirements.

Various steam pressure tests, 60 lb. to 250 lb. at 10-lb. intervals, one speed and all at atmospheric exhaust	20 tests
Repeat the above with various back pressures up to 30 lb., at 2-lb. intervals	15 tests
Total to date	300 tests
Repeat the foregoing for varying speeds, 1,000 to 4,000 r.p.m., at intervals of 100 r.p.m.	30 tests
Total to date, 300 × 30	9,000 tests
Repeat for ¾, ½, ¼ and no load	4 tests
Total to date, 9,000 × 4	36,000 tests
Repeat the same with from one to six nozzles, or five additional tests,	
Total to date, 36,000 × 5	180,000 tests

The reason for repeating all these tests is that the back-pressure correction varies with the speed on account of skin friction.

On the basis of twelve tests per working day this series would take 15,000 days, or fifty years. We have then only tackled one machine and no condensing tests.

The absurdity of the whole proposition is at once apparent, but these figures are given just to show how much the majority of guarantees given must necessarily be based on interpolation. Is it not surprising, therefore, that engineers have come as close as they have done when the foregoing is taken into consideration.

Let us see how the fan or blower situation stands. It is only during the last two or three years that any attempt whatever has been made to accurately determine the characteristics of different designs. The apparatus for fan testing is so expensive and cumbersome that it is out of the question to carry out tests on all sizes; we must therefore be satisfied with tests on one or two sizes only and interpolate the rest. Furthermore, the selling price of fans today leaves little margin for testing expenses. The saving grace in the fan situation has been the fact that the customer rarely knows how much air he wants, figures liberally and after installation has no means of finding out what the set is doing. Again, supposing the fan had been tested and gave excellent efficiency, how about the losses in the ducts and flues after installation?

The pump situation differs from the fan situation in the following respect: The pump maker unquestionably knows more about his apparatus than the fan maker, but if he falls down, he is so much more easily found out. Now let us compare the pump situation with that of the turbine. The general design of steam turbines has not varied much during the last eight or nine years, every turbine maker has a complete testing outfit, he has carried out innumerable tests, and these, together with the results of valuable reasearch work that have been published, put the turbine maker in a position to make a guarantee that is commercially accurate.

This much, however, cannot be said of the pump maker. To begin with, the designs have not yet been sufficiently standardized to get a reliable line on the performance of various types of impellers or diffusion vanes. Again, the testing equipment of the average pump plant is totally inadequate to carry out the tests

necessary for the formulation of accurate data. Pump makers naturally resent any arguments advanced along these lines, but the bare facts remain that the majority of guarantees must be based on partial-load tests or tests made on similar pumps installed and tested after installation. And yet in spite of all this, the pump maker almost invariably blames the turbine if the set falls down, and he has been known to go so far as to do this after himself witnessing a test on the turbine before shipment. Considering the small amount of data available and the comparatively few pumps that do fall down, great credit is due the pump designers. The point that is open to grave criticism, however, is their insistence in giving *absolute* guarantees, and sometimes absurdly high ones at that, when an intelligent engineering public, properly educated to the facts of the matter, cannot expect anything better than a close approximation.

Returning to the question of maximum rating of small turbines, in view of the foregoing, is it practicable at the present time? It would not appear so with the possible exception of generator work.

Assuming that all differences referred to could be satisfactorily adjusted between the turbine and pump or blower maker, would a maximum rated machine be satisfactory to the customer if he could not get full load because he had not allowed for a ten or fifteen pound drop in his steam line or he has a back pressure of two pounds instead of zero?

In conclusion, this whole subject is one that concerns the purchaser as much as the manufacturer, but it is up to the manufacturers first to get together and establish a code of allowances that must accompany each guarantee. The customers will then naturally fall into line when they see that the manufacturers are no longer charging them for contingencies that arise at the present time due to insignificant and immaterial discrepancies between the letter of the contract and the actual performance.

Hartford, Conn. W. J. A. London.

Heat Maladies of Marine Diesels

I have followed with interest the discussions in recent issues of *Power* under the heading of "Heat Maladies of Marine Diesels." Mr. Crowly's original article was timely and suggests a number of things that every Diesel-engine designer must consider carefully. It seems, however, that he has exaggerated the effect of temperature changes resulting from fluctuations in the power absorbed by the propeller of a ship in a rough sea. Relative to this condition he says:

"Indicator diagrams taken during the early strokes of an engine that has just been started show pressures up to 700 and 800 lb. *There is no danger in the pressures themselves, the engines are built to withstand them and to withstand much greater pressures; but concurrent with the increased pressures are increased temperatures, and they are potent for trouble.* To have these increased temperatures in the cylinders recurring every time the governor acts to prevent racing, perhaps two or three times a minute, not for a half-hour or an hour, but perhaps the whole day or several days, is a condition with which the land engine never has to contend."

The italics are mine. It is not clear on what basis Mr. Crowly assumed any such serious increase in temperatures under the given conditions, because it is merely a question of adding some of the heat at approximately constant volume instead of adding it all at practically constant pressure. Assuming the normal maximum pressure to be 500 lb. per sq.in., then if this maximum be increased 50 per cent. (to 750 lb.), it is easy to demonstrate thermodynamically that the maximum temperature cannot be increased by more than 7 or 8 per cent., depending on the conditions chosen. Of course, if the pressure goes still higher this percentage may be increased, but even if all the heat is added at constant volume it will not exceed 20 or 25 per cent., with a resulting maximum pressure, if not relieved, of 1,200 to 1,500 lb. per sq.in. This is probably the reason why "relief valves, though unusual in land practice, are orthodox fittings of the marine Diesel engine."

In view of these facts I cannot share Mr. Crowly's contention that the temperatures resulting from early combustion are more "potent for trouble" than the pressures. I admit that it is highly desirable to eliminate these higher temperatures and am aware that the history of internal-combustion engine development contains a record of more than one forward-looking individual who has pointed out the thermal advantage of adding heat at constant pressure instead of constant volume. However, gas and gasoline engines are still operating successfully.

Moreover, some of the larger gas engines in this country have a cylinder bore of 42 and 46 in., while the largest in Europe, I believe, is over 51 in. Of course, these are stationary, not marine engines. On the other hand, I am not aware of a single Diesel engine in continuous operation today with a cylinder bore exceeding 26 in.

In a large gas engine the normal maximum pressure is around 350 lb. and under abnormal conditions may possibly reach 500 or 600 lb., but no such excessive abnormal pressures as may be developed in a Diesel engine are possible.

It might be recalled here that a relief valve is not always effective in preventing an excessive pressure within a large internal-combustion engine cylinder, because the development of pressure may be so rapid that a small valve is insufficient and the inertia of a large valve may be so great that its movement is too slow.

Returning now to Mr. Crowly's specific statement that "there is no danger in the pressures themselves, the engines are built to withstand them and to withstand much greater pressures; but concurrent with the increased pressures are increased temperatures and these are potent for trouble," this might be restated so as to shift the argument somewhat as follows:

These temperatures themselves are not difficult to take care of, because cylinder walls rigid enough to provide suitable wearing surfaces for any desired size of cylinder can be effectively cooled by a water-jacket; but concurrent with these temperatures are excessive pressures, and these are potent for trouble in large cylinders. It is not temperatures alone, nor pressures alone, but the combination of the two that is serious, and it is largely beside the point at issue to try to draw a line of demarcation between them or to definitely state whether a 50 per cent. increase in pressure is more or less serious than a 10 per cent. increase in temperature.

The vital issue is, what can be done to eliminate or successfully control the excessive stresses in a Diesel engine incident to the tossing of a ship at sea? Unquestionably any Diesel-engine manufacturer, whether of two-

or four-stroke-cycle engines, would welcome gratefully any constructive information on this issue, but what is its relative importance in a two- and a four-stroke-cycle engine? In small engines there is little need of dwelling on structural differences, because the unit stresses in the walls may be kept within a safe margin in either case; but in large cylinders with thick walls the temperature and pressure stresses combined may become serious.

Fatigue of metal is no longer a myth, but the exact laws of fatigue under the combined action of both high pressures and temperatures are not yet definitely known. The life of a piece of metal strained intermittently to near its critical point of endurance is generally some function of both the number and frequency of applications. Therefore a greater margin of safety may be necessary in large two-stroke-cycle than in large four-stroke-cycle engines, or in other words, a somewhat larger cylinder may be used in a four-stroke-cycle engine with the same margin of endurance, other things being equal. Please notice this last phrase, because other things seldom are equal or can be made so, and this is why any sweeping statement as to the relative value of the two types can be of little or no value. The more complex structure of the cylinder head, combined with attendant increased stresses due to irregular cooling alone, in a four-stroke-cycle engine may offset completely the margin established in its favor above. Unfortunately, also, this question of fatigue in large cylinders cannot be determined on the testing floor nor by one or even three or four successful voyages across the Atlantic. Three or four years of continuous operation may be necessary to determine whether it is really serious.

There is another phase of this temperature problem, however, which tends to confirm Mr. Crowly's contention. Under normal conditions in a four-stroke-cycle engine the pressure in the cylinder at the beginning of compression is around 13 lb. per sq.in., while in a two-stroke-cycle engine it is around 16.5 lb. Also, the maximum temperature in a four-stroke-cycle engine seldom exceeds 3,000 deg. F.; therefore the increase in temperature due to combustion may be 2,000 deg. F. With cylinders of equal capacity and if the temperature at the beginning of compression is the same in each, then the increase in temperature during combustion in the two-stroke-cycle engine need be only $\frac{13}{16.5} \times 2{,}000 = 1{,}575$ deg. in order to develop practically the same mean effective pressure; that is, the maximum temperature would be 400 to 450 deg. less than in a four-stroke-cycle engine operating at the same mean effective pressure. Granting that this disparity in maximum temperature is necessary or desirable under normal conditions and assuming the two-stroke-cycle engine to be completely scavenged with air, what will happen at maximum load if the distribution of fuel becomes unequal so that one or more of the cylinders develop their full potential power?

Again, where the unit pressures are as high as they are in the Diesel engine the mere question of adequate distribution of lubricating oil at all times may be the determining factor between the two types. In engines of the trunk-piston type this problem becomes relatively more difficult with increase in both size and speed in two-stroke-cycle engines than in four-stroke-cycle engines. As the size is increased, the volume of hot exhaust gases that must pass over the walls and through ports at the bottom of the cylinder increases as the square of the diameter, while the circumference of the cylinder increases only as the first power of the diameter. In order, therefore, to keep the velocity of the exhaust gases within a reasonable limit, the length of the exhaust ports must be increased, as must also the general dimensions of the exhaust chamber around the cylinder. These, combined with the necessary increase in thickness of the cylinder walls, make adequate cooling of the webs between the exhaust ports increasingly difficult, and with this increased temperature of the walls over which the piston (carrying the connecting-rod thrust) must move comes increased dependence upon adequate lubrication at every instant during the operation of the engine. The effect of an increase in piston speed may be analyzed in much the same manner.

The chief lubrication difficulty at high speeds in two-stroke-cycle engines, however, seems to have been experienced in connection with the main bearings. There are two reasons why effective lubrication may be more difficult here than in the four-stroke-cycle engine, assuming the bearing surfaces to be equal in area in each case.

1. The power transmitted is nearly double.

2. The high pressures within the cylinder combined with the inertia of the reciprocating parts maintain at all times a positive pressure in one direction against the bearings, thereby increasing the difficulty of forcing oil under the pressure side of the bearing.

The first of these varies directly with the speed, and the second, or inertia, forces increase as the square of the speed. At high speeds, therefore, they make a formidable combination which must be met and overcome. Of course the effect of these inertia forces on lubrication in a two-stroke-cycle engine can be eliminated by making the engine double-acting, but this is not yet an accomplished feat, at least in a successful marine engine of large capacity. The use of a separate crosshead in a single-acting two-stroke-cycle engine simplifies the question of adequate lubrication of the piston, but if offered as a competitor with the trunk-piston four-stroke-cycle engine, it tends to reduce the margin of simplicity in construction.

Cambridge, Mass. M. R. WALFARD.

Steam-Bound Boiler-Feed Pump

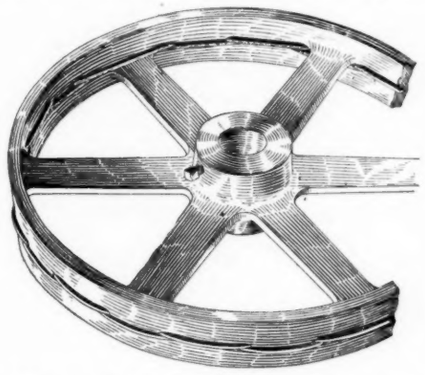
In regard to Mr. Gibson's article, "For Want of an Air Cock," in the issue of June 20, page 871, the following is the procedure with a steam-bound pump in the central station, where I am employed.

When a pump starts to "slam" and take short, jerky strokes, it is a sign that it is either steam- or air-bound, so shut that pump down by closing the steam throttle, leaving the rest of the valves alone, go to another pump, open the suction valve, open the exhaust and open the throttle, allowing the pump to take a few strokes (enough so that the water chamber is full of water). Then and not until then open the discharge valve slowly and give the pump more steam. The result is that whatever is in the feed line (air or steam) will be forced back into the boiler. If the discharge valve is opened before the pump has taken a few strokes and filled the water chamber, the steam will back up in the water chamber of that pump and it will also become steam-bound. To determine which of the check valves is in bad condition, use the method described by Mr. Gibson, or any opening in the common supply line will show if hot water is backing up.

Middletown, N. Y. THOMAS M. GRAY.

An Improvised Crown on a Flat Pulley

This simple and effective method was devised for keeping a high-speed belt on a large-diameter iron split pulley driven from a motor. Owing to both pulleys being flat, considerable difficulty was experienced in



AN IMPROVISED CROWN ON A FLAT PULLEY

keeping the belt in the center of the pulleys. In the large pulley ½-in. holes were drilled between each pair of spokes and in the center of the face of the pulley. Two belt laces were then threaded on alternate sides of the rim, forming a false crown, as shown in the accompanying illustration. Several methods were tried before this, but without success. I do not remember seeing this arrangement before, but consider it may be a useful kink for any of your readers who have a similar difficulty.

Harehills, Leeds, England. ARTHUR BINGHAM.

Repairing a Broken Cogwheel

The illustration shows how I repaired a broken cogwheel in what seems to me to be record time. The break occurred at 9:30 a.m., and the 100 or more hands about the plant expected to have a fine day fishing, but in 3 hr. they were called in to work again. Two bands were made to slip inside of the rim and were bolted to it between the cogs in countersunk holes, as shown. Bands were shrunk on the hub and a ⅜x½-in. ring was also fastened about midway of the length of the spokes.

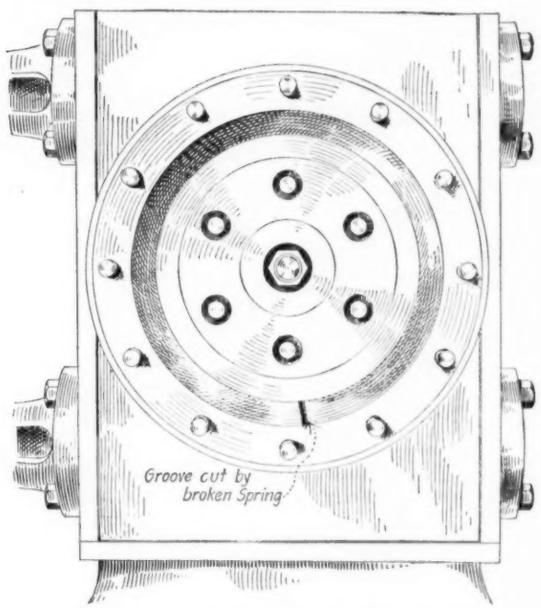
GEAR AFTER BEING REPAIRED

Shelby, Mich. RAY E. HAMILL.

Badly Cut Cylinder Repaired

The piston ring on a 16x42-in. Corliss engine broke and also one of the flat springs under the ring. One of the pieces of the broken spring was short enough to take almost a vertical position in the ring slot and form a cutting tool on the cylinder. Parts of the broken ring wedged against this cutting tool so as to hold it in one place. Before it was discovered, a groove ⅛ in. wide and ¼ in. deep had been cut the length of the cylinder, as shown in the illustration. The rest of the cylinder was practically untouched.

It was decided to try to repair the cylinder without reboring. Hard babbitt was first thought of for a filler, but with the tools at hand it was impossible to drill small



WHERE THE CYLINDER WAS GROOVED

anchor holes in the groove to hold the babbitt in place. It was then decided to use zinc for a filler. The groove was irregular in shape. A short cold chisel was made with a square cutting point ⅛ in. wide and shoulders to prevent the tool from going over ¼ in. deep. This tool was driven through the groove to square it. Of course several of these chisels were broken before the job was finished, and needless to say, the 16x42-in. cylinder was an awkward place in which to use a hammer and chisel, especially toward the front end.

After squaring up the groove, concentrated hydrochloric acid was poured into it to clean it. This was left in for five minutes and then washed out. Then a weakened solution—three parts water and one part acid—was used and left in the groove five minutes. This was thoroughly washed out with boiling water, and then the groove was dried.

Enough melted zinc was poured into the groove to a little more than fill it. This zinc was thoroughly peened

down with a ball peen hammer and most of the excess trimmed off with a flat chisel. The zinc was purposely left a little high so that the piston ring would cut a true seat. The engine was run for two weeks after repairing, and the cylinder head was then removed for inspection. The job was found to be in good condition. Six weeks later the cylinder head was again removed and the job inspected; it was found to be in good condition. How long the zinc will last with the engine in constant service is hard to say, but even if it does start to cut out, it will not be difficult to renew it.

Gillespie, Ill. F. F. JORGENSEN.

Piston Turned Top Side Down

A large duplex pump working a hydraulic elevator caused considerable trouble at times from noise and labored movement. Liberal lubrication did not seem to do much good, although some of the time, for some reason, the pump would behave first-rate.

I finally turned the pistons half-way round, and the noise disappeared for good. If the trouble was caused by the sharp edges worn on the piston rings scraping away the oil, why did it not act the same all the time and not come and go? H. K. WILSON.

New Bedford, Mass.

Operating Belt at High Speed

Some years ago I operated a 42-in. three-ply lapped and riveted belt at a little more than 92 mi. per hr. transmitting an average load of 700 hp.

The engine pulley was 16 ft. diameter, turning 165 r.p.m., driving to a 4-ft. pulley, and both had a ¾-in. crown. We had to keep the belt very tight to overcome centrifugal force, which at full speed would throw the belt edges away so that we could see a space between the belt and the pulley extending in 8 in. from the edge. This left not more than 26 in. width of belt in contact, yet it was delivering 700 hp.; but if not very tight it would throw farther from the pulley faces and slip. This drive ran satisfactorily for about sixteen years until discarded on account of changes in the mill requiring a larger engine.

Of course the wheel was very strongly built and was reinforced with a number of tie-rods, with turnbuckles, from hub to rim. C. A. GREEN.

Cleveland, Ohio.

Puzzling Pump Ailments

Two aggravating troubles came under my observation some time ago with two 6x10x12-in. vertical, simplex hotwell pumps controlled by a float in the hotwell. These pumps never operated properly, and a representative of the builder stayed with them over three months, but could not improve their working. The superintendent of the factory where they were made then came to investigate, but could not suggest anything, so the local force took up the hunt.

The pump barrels were set on the bedplate that had been faced to receive them, forming the lower head. A 3-in. plug screwed into the bedplate from the under side had worked loose and finally dropped out. As the bedplate had been set in cement, a new plug was inserted from its upper, or cylinder, side and no further trouble was experienced with this pump.

It was natural that we should look for the same trouble in the other pump, but we found it was not the same. This one had a porous cylinder casting and in tapping it a sand hole, large enough for the insertion of three fingers, was found connecting the two cylinder ends. It was repaired by filling the sand hole with lead, after which a ¼-in. cut was taken, with a boring bar, past this opening and a bronze ring turned to a driving fit and inserted in the end of cylinder.

We have since had no trouble with these pumps.

Fort Wayne, Ind. W. J. MAXWELL.

Two Good Operating Kinks

About two hours after cutting a boiler in on the line after making some repairs, a handhole gasket on the superheater blew out. It was necessary to have the boiler in service as soon as possible, as one boiler would not carry the load in the morning and the extra one was off for inspection. I let the steam down to zero and knocked in the handhole plate, and after leaving the boiler for an hour, it was still too hot to reach in and get the plate out by ordinary means, so I put on a canvas glove and put the nozzle of a ¾-in. hose inside the glove, then turned the cold water on. The "water jacket" around my hand enabled me to take the old plate out and put a new gasket on without burning my hand and also saved a couple of hours' time in getting the boiler back into service.

After having considerable difficulty keeping a three-valved trap tight because it was too large for the work it had to do, I removed the two outside valves and plugged the seats, making a one-valve trap of it. After that I had no more trouble with it, and it stayed tight as long as I was on the job. J. H. HONEY.

New York City.

Tally on Coal Conveyor

A coal-measuring device which I designed has been used on a 14-in. belt conveyor for the past three years. The conveyor takes screenings from a 1⅛-in. round-hole screen on the tipple of a coal mine and delivers it to the boiler house.

One set of the troughing idlers, which are on 5 ft. centers, was removed and replaced with a hardwood roller 1 ft. in circumference, placed about 2 in. below the belt when running empty. When the belt is loaded, it sags enough to bring it in contact with the wooden roller. A Veeder revolution counter registers the number of turns the roller makes, and the circumference being one foot, the revolutions represent the lineal feet of coal on the belt.

Running 100 ft. of screenings on the belt, we find it holds 8 lb. per ft. To get the amount of coal going to the boiler house we read the counter before starting and after the bunker is full, and multiply by 8; this gives the total weight delivered. The device cost about $10 to make.

Spring Valley, Ill. A. H. HALLADAY.

Where a Long Combustion Arch or horizontal baffling is used and is close to the fire, the furnace temperature melts the ash, causing the slag to fill up the air space in the grate, reducing the available boiler capacity. In a water-tube boiler, exposing the first row of tubes by putting the tile between the first and second rows will usually reduce the temperature sufficiently to avoid trouble.

Inquiries of General Interest

Value of Circular Mil—What is the value of a circular mil in inches? I. B.

A circular mil is the area of a circle that is 1 mil, or 0.001 in., in diameter, or 0.0000007854 sq.in., and is the standard unit of cross-sectional area of wires or other conductors used for carrying electric current.

Forward Stroke and Return Stroke—Which is the forward and which the return stroke of an engine? H. M.

The stroke of the piston from the head end to the crank end of the cylinder, as the working stroke of a trunk piston, is called the forward stroke, and the return stroke is made from the crank end to the head end.

Stopping Leaks in Ammonia Piping—What is a good way to temporarily stop leaks in ammonia piping? C. R. M.

Make a stiff paste of glycerin and litharge and apply to the leak. After the paste is applied, cover it with sheet rubber, held on by clamps. If the leaks are small enough, they may be easily and quickly closed with tin solder. Clean the metal where the solder is to be applied.

Cleaning Glass Gages—How can deposits and stains be removed from glass water gages? A. R. E.

Coatings held by deposits of grease can be removed with gasoline, and any cloudiness then remaining is generally removable with a weak solution of muriatic acid. If reddish-brown stains remain after cleaning a glass with gasoline, they are usually due to deposits of iron rust that can be removed with a solution of oxalic acid.

Universal Motors—What are universal electric motors, how are they constructed and for what are they used? L. W. C.

A universal motor is a series-connected machine that will operate on either alternating or direct current. The winding in it is practically the same as that used on a direct-current series motor. The polepieces as well as the armature core are laminated. They are built in small sizes, usually less than ¼ hp., and are used chiefly for driving household and office devices.

Allowance for Expansion of Buried Water Pipes—What allowance is to be made for expansion and contraction of buried water service pipes with screwed joints? P. H. C.

Ordinarily the variation of temperature of the water is not over 30 deg. F., and if the pipes are laid in ordinary temperatures of the atmosphere no allowance needs to be made for expansion or contraction. Allowing a coefficient of expansion of 0.000007 per deg. F., the variation in length, besides being gradual, rarely exceeds 0.0432 in. for a 16-ft. length of pipe. This is so small that the expansion and contraction are readily taken up at the joints or by lateral deflection of the pipe without serious damage.

Piston Speed of Small Pumps—Why do the builders of small direct-acting steam pumps limit the piston speed below that allowed for larger pumps? C. B. G.

The best working speed of a pump is mainly limited by the frictional resistance due to the size and form of the water passages, shocks occasioned by reversal of the pump, time required to open and seat the valves, and freedom of flow of the suction water so as to obtain "solid" water in the pump cylinder. In smaller pumps the shocks are less severe from handling smaller masses of water, the valves respond quicker and there may be more frequent reversals. But as the smaller pumps usually have shorter strokes, they must be run with a lower piston speed.

Tons of Pea Coal in Conical Pile—How many tons of 2,000 lb. of pea coal can be stored in a conical pile 80 ft. in diameter? R. L. K.

The angle of repose would be about 27 deg., or practically 0.5 vertical to one horizontal, and with that slope of the sides, a conical pile 80 ft. in diameter would be 20 ft. high. As the volume of a cone is equal to the area of the base multiplied by one-third of the altitude, the volume of such a conical pile would be $(80 \times 80 \times 0.7854) \times (20 \div 3) = 33{,}510$ cu.ft. The space occupied by a ton of pea coal varies with different kinds of coal, but for storage purposes it may be assumed as 37.5 cu.ft.; hence, in round figures, a conical pile 80 ft. in diameter would contain about 900 tons.

Absolute Pressure Readings for Boiler Gages—Why are not boiler pressure gages laid off to indicate absolute pressures? J. M. C.

The scale of a spring pressure gage may be laid off for indicating absolute pressures, but as the movement of the weighing spring is neutralized by the pressure of the atmosphere, which constantly varies for the same place and is different at different elevations, the readings generally would require correction. In operation of boilers and nearly all types of pressure vessels, it is usually of greatest importance to know the pressure that is present in excess of the pressure of the atmosphere, and it is of greater convenience both in construction and use of spring pressure gages to have their scales laid off in the usual manner, that is, for indications of pressures in excess of atmospheric pressure.

Obtaining Less Blowdown of Pop Safety Valve—A 4-in. pop safety valve was set to pop at 110 lb. and the blowdown was 5 lb. After setting the valve to pop at 155 lb., the blowdown was about 15 lb. How is the blowdown ring to be adjusted so as to reduce the amount of blowdown with the higher pop pressure? G. F.

To reduce the amount of blowdown, the ring should be so adjusted as to obtain freer escape for the steam between the upper edge of the ring and the flange that projects from the body of the valve above its seat. The escaping steam will then be of less assistance in holding the valve open and the valve will close for less blowdown of the boiler pressure. In most pop valves there is a screw cut on the blowdown ring, by means of which it may be raised to increase the amount of blowdown or lowered to obtain freer escape of the steam with less blowdown. A right-hand screw is generally employed to adjust the ring, and less blowdown is obtained by turning the ring so the side toward an observer is moved from right to left.

Three-Phase Motor Current Unbalanced—Can you explain why the current and watts are not the same normally in each phase of a three-phase motor? M. E. W.

If the voltage across each phase is the same, the current and watts in each phase of a three-phase motor should be the same. Probably the voltage is not balanced across each phase. This may be due to an unbalanced transmission line, or the ratio of the transformers used may not be exactly the same. Two wattmeters placed in a three-phase circuit to measure the motor load will not read the same due to the power factor. When two wattmeters are connected on a three-phase system in which the power factor is less than unity, the conditions are such that one meter will read fast while the other will read the same amount slow for a power factor above 0.5; at this point one meter stops. For power factors less than 0.5 one of the wattmeters will give a negative reading. Therefore the meter readings are added to obtain the watts transmitted when the power factor is above 0.5 and subtracted when less than 0.5.

Removal of Heat Radiated from Driers—What would be the best method of ventilating a workroom 20 and 40 by 60 ft. that is overheated by the presence of three steam dry-boxes which are equally spaced through the center of the room? J. O.

Over each dry-box it would be well to place a galvanized sheet-iron canopy or hood projecting about 2 ft. beyond the sides of the dry-box, with the sides of the canopies extended down as low as convenient for operation of the dry-boxes. For removal of the heated air, connect the top of each canopy with an exhaust fan, using liberal sizes of galvanized sheet-iron piping for the main pipe and branches, and have the exhaust fan discharge through a window or a special wall opening direct to the open air. The branch pipe from each drier should be provided with an adjustable gate or damper for equalizing the amount of air exhausted from the different driers. The air exhausted at the driers will be replaced by air drawn from the room, and ventilation of the room can be effected by admission of fresh air at such points as will obtain the desired distribution.

[Correspondents sending us inquiries should sign their communications with full names and post office addresses. This is necessary to guarantee the good faith of the communications and for the inquiries to receive attention.—Editor.]

Experience in Central Station Protective Features*

By N. L. Pollard and J. T. Lawson

SYNOPSIS—The protective devices and schemes discussed in this paper are as follows: Aluminum-cell arresters; arcing-ground suppressor; faulty cable localizer; cable testing; high-potential and high-frequency testing; reactors; relays; multi-recorder; insulation resistance recorder; resistance bulbs and thermocouples; dampers on air-blast transformers; coherer alarm devices; potential-indicating devices.

The entire territory served by the Public Service Electric Co. comprises three principal divisions—the Northern, Central and Southern—which include the more densely populated sections of the State of New Jersey. The Northern Division consists of eight generating stations having a com bined capacity of 148,000 kw. and feeding thirty-three substations. The Central Division consists of five generating stations having a combined capacity of 17,800 kw. and feeding thirteen substations. The Southern Division consists of four generating stations having a combined capacity of 32,000 kw. and feeding seventeen substations.

In the larger stations current is generated at 13,200 volts three-phase, both 25 and 60 cycles, and in most cases is distributed at that voltage between the various stations and substations, through 260 miles of underground cable and 425 miles of overhead lines.

In the smaller stations current is generated at 2,400 volts two-phase 60 cycles and distributed locally at that voltage. That part of the current not consumed locally is stepped up to 13,200 volts three-phase by means of Scott-connected transformers.

In certain sections the loads have increased so rapidly that it has been found necessary to change some of the transmission lines to 26,400 volts, and to install step-up and step-down transformers with a ratio of two to one.

The method of operation is to run all stations in multiple, which necessarily means that the largest and most economical stations deliver the most output. The older and smaller stations are used at peak periods and as stand-by stations.

Six years ago, in the Northern Division, there were too many cable failures in relation to the mileage. The total number of line and cable interruptions since 1913 has decreased, while the mileage has increased. This was brought about by eliminating as rapidly as possible all equipment that was proved defective and by the use of such safety devices and connection schemes as are described later.

ALUMINUM-CELL ARRESTERS AND ARCING-GROUND SUPPRESSOR

While the installation of aluminum arresters has made the apparatus trouble disappear almost completely and the failures of cables have decreased, it has, as might be expected, not given all the desired protection to the cables. The reason for this is probably due to the fact that the surges on the cables are in the form of distributed charges and the potentials rise locally in the cables entirely beyond the protective influence of the arrester.

Upon analyzing these cable failures, it was noticed that the majority of them were caused by faults to ground which later developed into short-circuits. At this time due consideration was given to the question of grounding the neutral of the system, but after taking all the factors of the problem into account, including continuity of service, etc., the idea of grounding the neutral was abandoned and it was decided that, as far as the system was concerned, the best remedy was the arcing-ground suppressor.

An arcing-ground suppressor was installed in the largest generating station and numerous aluminum-cell arresters on different parts of the system, in the belief that they would be the remedy best adapted to meet the requirements. The arcing-ground suppressor has been in service about five years, and the records show that it has operated in every case where a fault to ground occurred, by extinguishing the arc and preventing an interruption to service.

In addition to preventing service interruption caused by single-phase grounds, it has the great additional advantage of protecting life. There are at least six cases on record where actual contact with the 13,200-volt wires was made by the workmen, which did not result fatally. There is no question but that each of these cases would have been fatal if the suppressor had not been in use. This conclusion is based on experience with the rest of the system not equipped with a ground suppressor and also on the system before the suppressor was installed.

The arcing-ground suppressor consists of three single-pole independent motor-operated oil switches, electrically and mechanically interlocked, to prevent more than one operating at the same time. Each switch is connected to ground on one side and to the bus on the other. The suppressor is controlled by a balanced three-phase potential relay, which remains inactive while the system is balanced, but when unbalanced, due to a ground on one phase, it operates the corresponding phase of the suppressor, which in turn grounds the same phase of the bus; thus shunting the current and extinguishing the arc. In cases of short-circuit, an extra precaution is taken to prevent possible operation of the suppressor by the addition of an overload relay which opens the control circuit of the suppressor.

FAULTY CABLE LOCALIZER AND CABLE TESTING

Working in conjunction with the suppressor is a device known as the faulty cable localizer, which serves the purpose of indicating the particular feeder on which an arc to ground occurs.

This device consists of a relay connected in series with the neutral of the feeder-current transformers. When a ground occurs, the secondary current of the transformers becomes unbalanced and causes the relay to operate. This in turn rings a bell and lights a pilot lamp which indicates the faulty feeder. It takes about 0.15 of a second for the localizer and 0.3 of a second for the suppressor to operate.

There is one record of a 15-minute interruption to the service in the Northern Division caused by a cable end bell short-circuiting on an armature lead of a generator in one of the stations, before the suppressor and localizer were put in service; the same thing occurred again and the situation was handled in such a manner that no one outside of the power station was the wiser. A number of instances are on record where a ground has occurred on substation busses without an interruption to service.

As a further means of reducing cable troubles to a minimum a careful investigation was made of the possible causes of failure. A thorough study was made as to the best method of making cable joints and particular attention was given to such factors as, (1) favorable weather conditions, (2) elimination of impurities, air and moisture, (3) application of insulating varnish between each layer of tape, (4) careful and even winding of tape, (5) improving the human element.

Aside from installation precautions, we have made it a practice to inspect cable during the process of manufacture and to see that the cable specifications are strictly adhered to.

All cables are tested with 26,000-volts for three minutes before putting them in service, but we do not make a practice of testing them periodically. In case there are indications of trouble on a cable while in operation, it is cut out of service, given a test of 26,000 volts for three minutes and returned to service in case no failure occurs.

In our opinion a 13,200-volt cable should not be tested at a voltage higher than 26,000, since our experience has shown that too high a testing voltage oftens weakens the insulation at some point, which weakness finally manifests itself in a complete breakdown, even under normal operating voltage.

HIGH-POTENTIAL AND HIGH-FREQUENCY TESTS

Our line insulators used a number of years ago consisted of many different types, no one type having been standardized. The old insulators not only spilled over during trouble, but also frequently became punctured. The insulator-creepage surface was then increased, and the insulators tested with high-potential 60-cycle current, but without satisfactory results.

About two years ago we started testing with high-frequency and were soon convinced from the results obtained

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that the insulators were not capable of standing this test. Various types of insulators were then experimented upon and a design finally adopted having ratio of puncture to flash-over of about 2 to 1. Now, every insulator before being placed in stock or used on the lines is given a 15-second high-frequency test. Although these insulators are previously tested at 60 cycles, about 2 per cent. fail to pass the high-frequency test.

At the present time, generators of such large capacity are being used that it is necessary to protect them from the effects of disastrous short-circuits by means of internal or external reactance. Where several generators are operated in parallel, it is essential that bus-tie or bus-section reactors be used in order to prevent the combined capacity of all the generators from feeding into any one point of short-circuit. Bus-section reactors will limit the current on any one section, but take up valuable space in the bus, complicate the connections, and in cases where parallel feeders are connected to different sections of the bus, it is difficult to obtain balanced currents in these feeders. The use of tie-bus reactors has none of the disadvantages of the bus-section reactors and does limit the amount of short-circuit current on any one section to any value desired, depending upon the amount of reactance used.

RELAYS FOR SELECTIVE SWITCHING

It is obvious, that practically all interruptions in a transmission system will be minimized if faults are easily and quickly removed before they have had time to cause serious trouble.

In order to determine the proper setting of instantaneous overload, inverse time-limit and inverse definite time-limit relays which are more commonly used on a system of distribution, it is necessary to know the characteristics of the system as well as the characteristics of the generators, automatic apparatus, circuit-breakers, regulators, etc., before anything can be done along these lines, and before the time elements of relays can be adjusted, the following information is necessary:

1. The instantaneous short-circuit value of current through each conductor to which the relays may be applied.
2. The sustained short-circuit value of current through each conductor.
3. The time in changing from 1 to 2.
4. The time required for various automatic circuit-breakers to open the circuit after application of current through the trip coils.
5. The safe current-opening characteristics of various circuit-breakers.
6. The time characteristics of the various relays.
7. The probability and amount of flow of energy in the case of circuits operating in parallel.

Assuming for the moment that the seven items listed are known and an attempt is made to accurately time relays for selective action, the accuracy of current values required to trip a relay can be left as a matter of small concern, for when a short-circuit occurs, the current setting of relays is usually exceeded by at least several hundred per cent. This applies equally well to inverse time or definite time relays. It is an impossibility to get selective setting with the inverse time-limit relay because it has the unfortunate characteristic of being instantaneous with a heavy overload. This characteristic prevents the use of this type of relay for selective action, because a short-circuit of sufficient magnitude will cause all the relays from the fault to the source of supply to become instantaneous, and thus any or all of the relays are liable to trip out instead of only the ones nearest the fault.

Better results can be obtained by the use of an inverse definite time-limit relay, the time setting of which depends upon the damping action of a permanent magnet on an aluminum disk, giving as great a degree of accuracy as can be obtained in the calibration of watt-hour meters used in connection with a torque compensator.

This type of relay is more accurate and allows closer setting than any other type, and when once set, remains the same, but in an extensive system the results obtained from its use are nevertheless disappointing. The reason for this is simple, when it is considered that the time interval between successive circuit-breakers should be equal to the time taken for the circuit-breaker to open its arc, plus a margin of safety to include variation, and if there are several feeder sections in series and the trouble should be near the generating end, the short-circuit may not be cleared for several seconds, which means, of course, a loss of all synchronous load on the system, or, in other words, a complete interruption to the service.

From the foregoing it can be seen that, irrespective of the character of the distribution system, it is practically impossible to isolate faulty cables or lines by the relays most commonly used, and therefore the records of most operating companies show that faulty feeders usually interrupt a large area and a great many more consumers than is necessary.

With the relays more commonly used, the most satisfactory results are obtained by using them with a radial distribution system; or, in other words, most operating companies get the best results by adapting their systems to meet the faults in the relay rather than design a relay for a minimum cable outlay.

In connection with the seven tie feeders between City Dock and Marion so much trouble was experienced owing to the lack of a selective action of the relays, that a new type of relay known as the "balanced selective relay" was finally installed at each end of these feeders.

The relay coils are connected in series with current transformers of the tie lines. This causes the current in each coil to be proportional to the current in the cable to which it is connected.

Only a single phase of each cable is taken care of by a single relay, and there are as many coils on the relay as there are cables. For instance, with seven cables on a three-phase system there are three relays with seven coils per relay.

Each coil on each relay is mechanically balanced against each other coil. The effect of this is that any strong coil can overcome any weak coil. In case a short-circuit occurs on one cable, the current in the faulty cable will of course be greater than the current in any other cable. Therefore, the coil connected to the current transformer on the faulty cable will have the strongest pull and will close the tripping circuit of this coil switch.

However, if none of the parallel cables are short-circuited, but trouble occurs on some radial cable, the relay will remain balanced and none of the good cables will be erroneously tripped out. The relays operate simultaneously at both ends of a faulty cable, and their operation is not affected by fluctuation in voltage due to trouble or change of phase angle between voltage and current. With this particular type of relay, it is necessary to have not less than three feeders in service, in order to secure perfect operation under all conditions. It should be further borne in mind that reactance coils are installed on both ends of the tie feeders in order to secure the best selective action of the relays. It is questionable whether the relays would operate as satisfactorily in all cases without the aid of these coils. The successful operation of these relays is limited to three or more feeders, therefore they cannot be used on other parts of our system where there are but two parallel cables between stations.

Since the installation of the balanced selective relays, we have had no case of cable failure on the tie feeders where the relays have not operated properly, and there have been more than fifty cases of trouble.

Our operating conditions are such that at times there are but two tie feeders in service, therefore on these occasions it is necessary to disconnect the balanced selective relays and depend on overload protection.

Judging from past experience with different types of relays, a pilot-wire relay would seem to be the best one to adopt if it were not for the trouble and expense entailed by the necessity of the pilot wires. Therefore some scheme based on the pilot-wire principle, having all the advantages and none of its disadvantages, would be an ideal one for our interconnected system.

Our investigation shows that the nearest substitute for the pilot-wire scheme and one that has practically all the advantages is the split-conductor principle. During the coming year we expect to make several installations of this character.

MULTI-RECORDER AND INSULATION-RESISTANCE RECORDER

The multi-recorder is a device for recording the time to the fraction of a second of the sequence of action of oil switches, circuit-breakers, potential-indicating devices and aluminum cell arresters. A record of this kind is invaluable to the station man in analyzing troubles or ordinary switching changes.

In order to have complete information on the performance of station apparatus under all conditions, it is highly desirable to have records of the closing and opening of circuit-breakers, operation of lightning arresters, appearance and duration of high voltage in the lines and the occurrence of grounds and short-circuits.

The insulation-resistance recorder is an instrument that gives a daily graphic record of the insulation resistance of the system. The results obtained so far from the use of this instrument have been rather disappointing, owing to the fact that there are so many old insulators of different insulation characteristics. When the insulators are all changed so that they will have the same characteristics throughout the system, we can, by establishing a point on the chart which might

be called "dangerous," but which is well above the breakdown point, give the operator an opportunity to report when the insulation of the system reaches this value. By isolating the transmission circuits one at a time, the line in question can be removed from the system and later, by high-potential testing or other methods, the bad insulators can be located and removed from the line.

In using this instrument on lines where the insulators have been standardized, very good results have been obtained. Its use has also shown the need of increasing the insulating qualities of porcelain, which is something that we did not know before.

RESISTANCE BULBS AND THERMOCOUPLES

For several years it was our practice to install resistance bulbs in the windings of the largest turbo-generators and synchronous motor-generator sets, in order to know at all times the temperature of these machines. The results obtained by their use so far have not met with expectations.

The resistance bulbs furnished today are somewhat more substantial than the earlier type, but are still too frail to stand much hard usage. In the windings of our more recent motor-generator sets thermocouples have been installed which we believe will give better service than resistance bulbs, as they are less liable to mechanical injury.

The temperature indicator has been found very useful in determining when a machine needs a thorough cleaning. Thus a possible burnout is prevented which otherwise could not be foretold.

It is our opinion that wherever possible thermocouples should be installed at the hottest points in the windings of all large generators or motors.

DAMPERS ON AIR-BLAST TRANSFORMERS

In most of our stations where air-blast transformers are installed, a common air chamber is utilized for all the transformers, and we have found this method of air supply more economical than to use a separate and independent blower for each transformer. Each transformer is equipped with a top and bottom air damper, so that in case of fire in the transformer the dampers may be closed, thus shutting off the air supply and smothering the fire.

We strongly recommend top and bottom dampers on all air-blast transformers, especially where a common air chamber is used. If the transformers are not equipped with both top and bottom dampers and a fire starts in one of the transformers, it is necessary to cut off the entire air supply for a considerable time, depending upon how long it takes to put out the fire. This might compel the station man to disconnect the other transformers from the service, thus resulting in a complete interruption.

COHERER ALARM DEVICE

The coherer alarm device is used to register predetermined voltage rises on the transmission line itself, across reactance coils or on aluminum-cell lightning arresters. In order to get a permanent record of its action, it can be connected to a relay which will make a record on a multi-recorder. This device has been found very convenient to register the discharges of our aluminum-cell lightning arresters.

Previous to the installation of this device it had been the custom to install with each aluminum-cell arrester a discharge recorder which has a continuously moving paper punctured by the discharges. We finally came to the conclusion that while these recorders gave us much desired information, the continual replacement of paper record rolls was expensive and troublesome.

POTENTIAL-INDICATING DEVICES

Electrostatic potential-indicating devices are used to indicate potential on feeder circuits. The instrument is connected to the middle point of two strain insulators in series, which are connected between each live wire and the ground. The displacement current through the insulators is sufficient to operate the instrument.

Another method of obtaining the voltage indications is to install potential transformers whose secondaries are connected to indicating lamps or voltmeters. However, this means is rather expensive and the required space for potential transformers is not always available.

It would therefore seem that the potential-indicating device now on the market might fill a long-felt want.

A Committee of Citizens of Lexington, Ky., where an anti-smoke ordinance is before the City Commission, has obtained from Mayor J. C. Rogers a promise that the pending ordinance will not be pushed at this time. The business men and others operating furnaces or boiler plants generating smoke wish further time to study the matter and to agree on a measure that they consider fair.

Annual Convention N. A. S E.

From Sept. 11 to 16 the National Association of Stationary Engineers will hold its thirty-fourth annual convention in Minneapolis. Headquarters will be the West Hotel, and the exhibits will be displayed in the armory, a five-minute car ride from the hotel. An excellent program has been prepared, the big day being Thursday, when all will go to the South St. Paul picnic grounds and participate in a Western beef barbecue and field-day sports. Following is an outline of the official program:

Monday, Sept. 11—Official registration of national delegates, members and ladies at West Hotel. During the day the local committee, accompanied by a band, will meet arriving delegations at railway stations and escort them to headquarters. At 8 p.m., official opening of the mechanical exposition at the Minneapolis armory, Kenwood Parkway, Chairman Williams, of the local committee, presiding; address of welcome by Lane Thompson, president of the National Exhibitors' Association; response by National President W. H. Damon. The exposition hall will be open throughout the convention from 9 a.m. to 10 p.m. At 8:45 p.m., reception of national officers, ladies, delegates and friends at the armory; 9:15 p.m., informal dance in convention hall, music by Rossiter's First Regiment Band.

OPENING SESSION OF CONVENTION

Tuesday, Sept. 12—At 9 a.m. officers, delegates and visitors will assemble at the hotel headquarters, and at 9:15 a.m. lines will be formed for the purpose of marching to the convention hall, escorted by the First Regiment Band and a squad of mounted police. At 10 a.m. opening exercises at the convention hall, J. M. Williams, chairman of the local committee, presiding; singing of "America" in unison; invocation by Rev. L. R. S. Ferguson, of St. Paul N. A. S. E. No. 7; address of welcome on behalf of the State of Minnesota by Hon. J. A. A. Burnquist, governor; response by National President Walter H. Damon; address of welcome on behalf of the City of Minneapolis by Hon. Wallace G. Nye, mayor; response by National Vice-President John Kerley; address of welcome on behalf of the Minneapolis Civic and Commerce Association by L. F. Blyer; response by F. L. Ray, past national president; official opening of the business sessions of the convention by President Damon; appointment of convention committees. At the conclusion of this session the official photograph will be taken in front of the armory. At 2 p.m., business session at the convention hall in the armory; opening exercises of the National Ladies' Auxiliary, Colonial Room, West Hotel; 3 p.m., lecture by Prof. C. F. Shoop, University of Minnesota, subject to be announced later; 4:30 p.m., annual meeting of the life and accident department in the convention hall, side room No. 2, armory; 8 p.m., theater party to the ladies at Shubert Theater; 8 p.m., officers, delegates and members will assemble in the exhibit hall at the armory for educational purposes along mechanical lines.

Wednesday, Sept. 13—At 9:30 a.m., business session of the national convention; 9:30 a.m., business session of the Ladies' Auxiliary, Colonial Room, West Hotel; 11 a.m., lecture by Prof. J. J. Flather, University of Minnesota, subject "The Engineer"; 2 p.m., business session of the national convention; 2:30 p.m., session of the Ladies' Auxiliary, Colonial Room, West Hotel; 3 p.m., lecture by H. F. Mueller, past state president and chief engineer of the Washburn-Crosby Co.; 3:30 p.m., automobile ride for ladies who have registered and have the official badge, sightseeing of city, lakes and boulevards, to terminate at the Washburn-Crosby mills; 4 p.m., officers, delegates, exhibitors and friends are invited to adjourn and visit the Washburn-Crosby flour mills, the largest in the world, joining the ladies there; 8 p.m., all are invited to the exhibit hall; 11 p.m., stag theater party for men only at the Gayety Theater.

Thursday, Sept. 14—At 9:30 a.m., business session of the convention; 9:30 a.m., session of the Ladies' Auxiliary, Colonial Room, West Hotel; 11:30 a.m., memorial services of the national convention; 12:30 p.m., Western beef barbecue, with all the accessories, and field day at South St. Paul picnic grounds; addresses will be made on the grounds by ex-Mayor Keller and Mayor Clark; response by National President Damon and Vice-President Kerley; 8 p.m., annual exhibitors' entertainment will be held in the convention hall at the armory.

Friday, Sept. 15—At 9:30 a.m., session of national convention at the armory; 9:30 a.m., final session of the Ladies' Auxiliary at West Hotel; 8:30 p.m., grand ball at hotel headquarters, West Hotel, music by Rossiter's Orchestra, dancing 8:30 to 12 p.m.

The National Exhibitors' Association will hold its annual meeting Monday, Sept. 11, in the convention-hall committee room at 4:30 p.m. The second meeting will be on Thursday, Sept. 14, at 4:30 p.m., same hall.

Measuring Water with a Weir Meter*

By E. G. Bailey†

A V-notch or rectangular weir offers many advantages for measuring water at or near atmospheric pressure. The flow curve is such that greater accuracy is readily obtained over a wider range in rate of flow than with the venturi or pitot tubes. From an accurate knowledge of the flow curve for any given shape of weir notch, it is possible to use the level of the water above the notch to continuously record the rate of flow and integrate it so that the total may be read from time to time. The nature of the flow curve of a V-notch weir, as shown in Fig. 1, brings out clearly the point that at low rates of flow there is proportionately a great deal of head, or power, to operate any float-driven mechanism. The converse is true at the higher heads, where a very slight change in head means a much greater change in the rate of flow.

The accuracy of the V-notch has never been questioned, but the points of primary importance are to accurately measure the head and to be sure that this measurement is referred back to the true zero of the notch. These points are

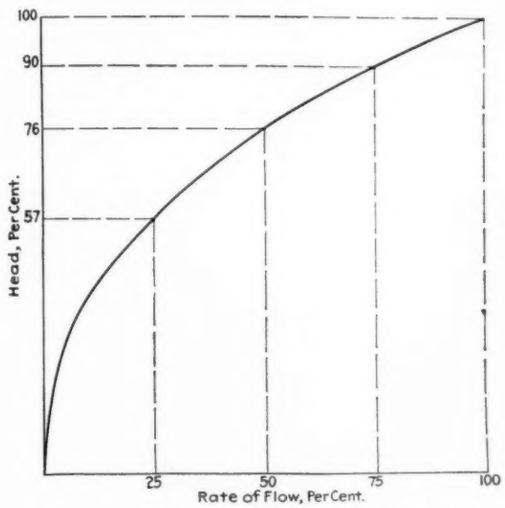

FIG. 1. FLOW CURVE OF V-NOTCH WEIR

difficult enough when using hook gages with micrometer screws in a hydraulic-engineering laboratory, and even more so when continuous recording and integrating mechanism is used to measure the rate and total flow.

The zero level can be readily obtained in test or calibration work if a permanent hook-gage point is riveted to the weir plate as shown in Fig. 4. This can be accurately located in the machine shop and is correct by construction.

It is preferable to have records on uniformly graduated charts, and in order to use the best form of integrating devices it is essential to have a motion directly proportional to the rate of flow instead of proportional to the head. There are many ways in which this may be accomplished with more or less accuracy, but the meter with the simplest mechanical construction gives the best satisfaction and continued accuracy over a long period of time.

The mechanism shown in Fig. 2 operates on the buoyancy principle. The displacing members are exactly balanced and weighted so that they are heavier than water and do not act as floats. If they were of equal and uniform cross-sectional area, the rising of the water level would produce no motion, but would merely cause them to become lighter on their knife-edge supports by an amount equal to the weight of water displaced. But by properly shaping these spun-copper displacing members, a powerful motion that is directly proportional to the actual flow is obtained directly from the mainshaft itself.

In Fig. 2 diagram A shows the positions of the two displacing members at zero flow, with their bottoms on the surface of the water at a level with the lowest point of a V-notch. Diagram B shows the positions of the displacing

*Abstract of a paper read before the June meeting of the Ohio Society of Mechanical, Electrical and Steam Engineers at Cleveland.

†President, Bailey Meter Co., Boston, Mass.

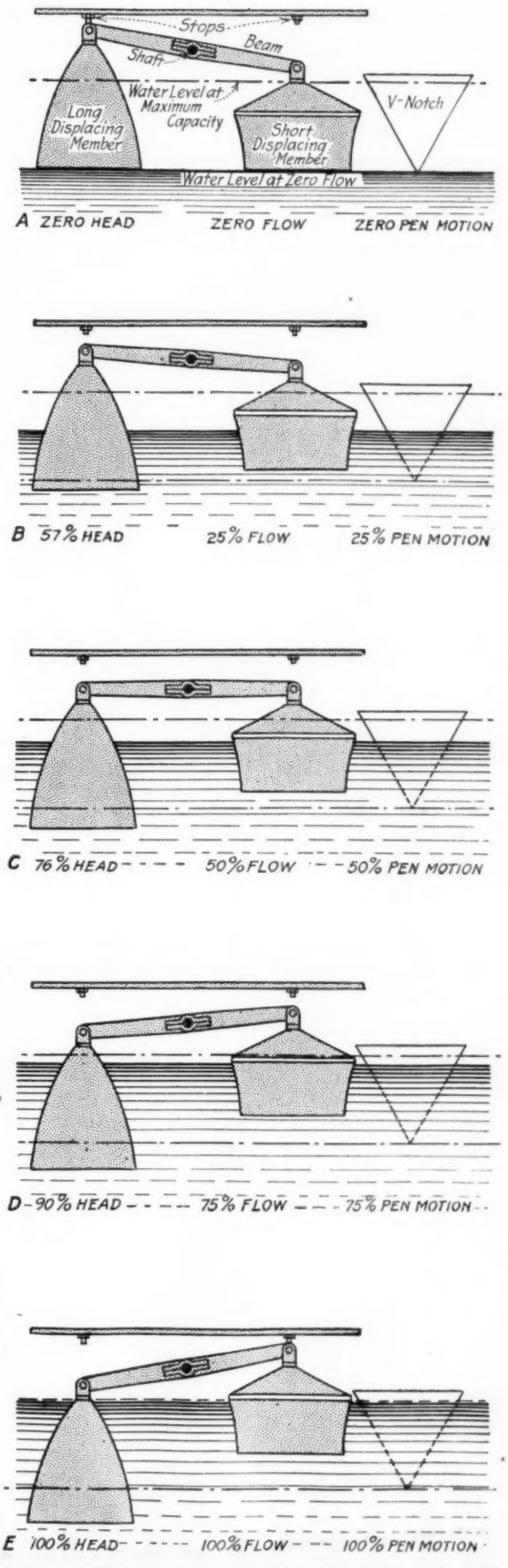
Stops
Beam
Shaft
Water Level at Maximum Capacity
V-Notch
Long Displacing Member
Short Displacing Member
Water Level at Zero Flow
A ZERO HEAD ZERO FLOW ZERO PEN MOTION
B 57% HEAD 25% FLOW 25% PEN MOTION
C 76% HEAD 50% FLOW 50% PEN MOTION
D 90% HEAD 75% FLOW 75% PEN MOTION
E 100% HEAD 100% FLOW 100% PEN MOTION

FIG. 2. SHOWING HOW UNEQUAL INCREMENTS OF HEAD ARE TRANSLATED INTO EQUAL INCREMENTS OF FLOW

members when the rate of flow is 25 per cent. of the maximum capacity of the meter. It is noted that the level of the water has increased to 57 per cent. of the total range in head. The motion of the displacing members, and also the recorder pen, has not been relatively as much as the change in head, but it is directly proportional to the rate of flow. The total motion of each displacing member is of course less than the total change in water level. The maximum head is about 10 in., and the corresponding motion of the displacing members is 4 in.

Diagrams C, D and E show the positions of the displacing members when the V-notch is discharging 50 per cent., 75

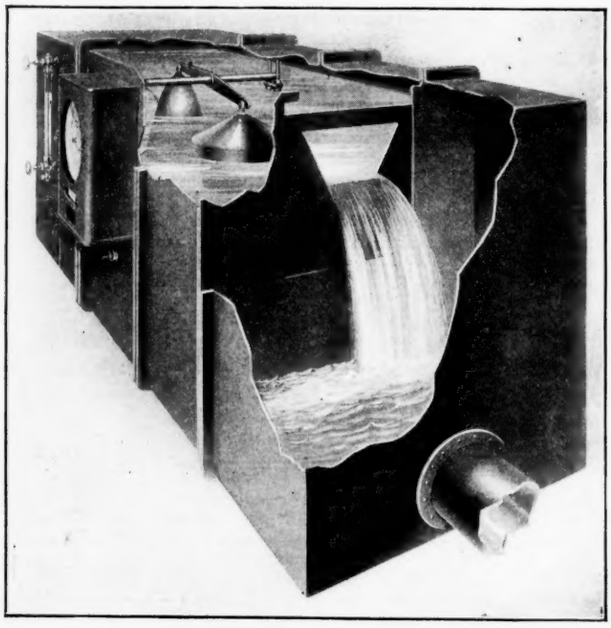

FIG. 3. VIEW OF METER IN PART SECTION

per cent. and 100 per cent. of its maximum capacity respectively. The last 25 per cent. in the rate of flow is effected by only a 10 per cent. increase in the head. In all cases the volumes of the two displacing members below the surface of the water are equal.

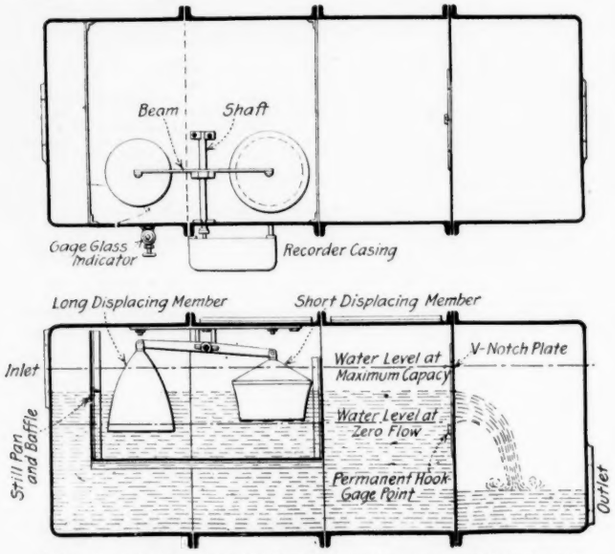

FIG. 4. PLAN AND ELEVATION OF BAILEY METER

The characteristic of the flow curve of a V-notch weir is the basis for the often-repeated argument of its great accuracy at the low rates of flow. But upon closer analysis it is found that this argument is a boomerang when considered in relation to the higher rates of flow, unless the recording mechanism is properly designed. An error of a few hundredths of an inch in a hook gage or straight float that rises directly with the water level may mean as much as 5 per cent. error in the flow reading at the higher rates.

The design of this weir meter obviates the possibility of such an error, for the relative positions of the two displacing members are made right by construction, and no changes can possibly take place to alter them. The pen and integrator can be set to their true zero so long as the water level is within ½-in. of the bottom of the notch, and there will be no error at the higher readings.

The working arrangement of these displacing members is shown in Figs. 3 and 4. The beam that carries the displacing members on monel-metal knife-edges is fastened to a shaft supported on monel-metal knife-edges. The end of this shaft extends through a pressure-tight bearing and carries the recorder pen and integrator driving directly on its outer end.

Boiler Explosions in Great Britain in 1915

The annul report of the Committee of Management of the Manchester Steam Users' Association says that the total number of boilers under inspection by that company on Dec. 31 of last year was 10,793. (The total number of boilers inspected in Massachusetts during the same year was 26,443.) After sixty-one years of immunity the association had met its first fatality during the year; in the scalding to death of an attendant by the bursting of a tube in a water-tube boiler. Outside of its ranks the association had noted throughout the United Kingdom during the year the occurrence of 48 explosions, killing 22 persons and injuring 54 others. Of these 19, killing 12 persons and injuring 21 others, may be termed boiler explosions proper, while the remaining 29, killing 10 persons and injuring 33 others, may be classed as miscellaneous explosions, such as those arising from steam pipes, stop valves, stills, drying cylinders and bakers' ovens.

Uniformity of Boiler Laws

The "News" has previously referred to the need of adequate boiler laws in the United States, the almost criminal record of 1914, when 300 persons were killed and 476 injured, having been cited as a compelling reason for the immediate attention of the state legislatures to this vital question.

It would seem, of course, at first glance that the method by which this desirable legislation could be most quickly and effectively secured would be the passage of a Federal law to govern the construction of boilers. But the Federal authority cannot extend within the boundaries of the individual states unless issues of interstate commerce are involved. The extension is prevented by the doctrine of state rights, and it is the more to be regretted because of the great amount of good that even the limited operation of Federal control of the problem has already brought about. For instance, at the present time there are in force Federal laws exercising control over boilers on navigable waters and on locomotives. As a result, to take the case of the locomotives, the annual number of accidents has been reduced in four years from 856 to 91, according to the report of the chief inspector of locomotives for the fiscal year ending June 30, 1915. The number of persons killed each year has come down from 91 to 13 and the number of persons injured from 1,005 to 467. In the fiscal year 1915 there were 32,666 locomotive boilers found defective and 2,027 were ordered out of service. These figures show beyond doubt that it is in legislation that the remedy for boiler accidents in general is to be sought.

But because of interference with state rights, the extension of the Federal jurisdiction, however effective it might be, is not possible. The duty, therefore, devolves upon the separate states. Here, however, a new difficulty arises. If each state acts independently in the matter, there can be no uniformity in the rules. And since the requirements governing safety in the construction of boilers must be uniform to be effective and harmonious, it is almost a necessity that they emanate from the same source.

This consideration left the advocates of effective and adequate boiler laws in a rather difficult predicament, from which there seemed to be no escape. Furthermore, the difficulty of the predicament was being annually enhanced by the passage of separate laws in separate states regarding the construction of boilers. A condition of chaos was being fast approached. At this juncture the American Society of Mechanical Engineers, recognizing the critical status of the problem and perceiving that a new direction must be given to the movement if it was to be saved from ultimate collapse into confusion, appointed a committee to draft a set of rules that might be a guide to those state legislatures which in the future took up the matter of boiler laws. This committee at

once set to work to draw up standard specifications of boiler construction, designed to make the best boiler possible and to eliminate, as far as it was humanly possible, the chances of explosions. Data were gathered from corresponding societies and engineering organizations throughout the world. All previously used rules for the construction of boilers and all legislation that had been passed on the subject came before the committee for critical discussion. There was finally published a set of rules which was circulated broadcast throughout the world—to the best engineers of all countries—urging their criticism and suggestions; and the rules were then revised in the light of these suggestions and criticisms.

These rules are now known officially as the Boiler Code of the American Society of Mechanical Engineers and are published in book form. The Code is indorsed and advocated by such bodies as the American Boilermakers' Association, the National Tubular Boiler Manufacturers' Association, the Association of Master Boiler Builders, the National Association of Thresher Manufacturers, the National Electric Light Association, the National Boiler and Radiator Manufacturers' Association, besides a multitude of locomotive manufacturers, hoisting engine manufacturers and manufacturers of steam shovels and other industrial tools. Practically all the insurance companies that write boiler insurance are also behind the movement for the adoption of the Code. At the recent meeting of the Texas Cotton Seed Crushers' Association, moreover, to bring the matter nearer home, the following resolution, introduced by P. S. Grogan, was unanimously adopted:

Whereas, The American Society of Mechanical Engineers having prepared a satisfactory code for the standardization of boiler construction to promote safety of human life and economy in operation, it is hereby

Resolved, That the Texas Cotton Seed Crushers' Association, in convention assembled, approves the said boiler code and recommends its adoption by the State of Texas.

Since the formulation of the Code by the American Society of Mechanical Engineers, several states have adopted it. According to C. W. Obert, secretary of the boiler code committee of the society, the States of Ohio, Pennsylvania and California have accepted the Code unconditionally and have embodied it as a whole in their statutes. In Wisconsin and Indiana the Code is acceptable as an alternative to the existing requirements. In the remaining states of the Union there are practically no laws governing the construction of boilers.

It is to be hoped that state legislatures all over the country will soon take official cognizance of the annual preventable loss of life resulting from the lack of boiler laws—and, having done so, will follow the good example of California, Pennsylvania and Ohio by adopting the Code of the society instead of one of their own formulation. This would secure the much-desired uniformity and would, moreover, be a move of economy because of the fact that all the disputed points in connection with the drafting of these rules have been thought out and determined upon by experts; and any attempt at selecting another set of rules would imply a duplication of labor and expense, with the probable arrival at the same conclusion if a like amount of consideration be given to it. The "News" especially hopes that Texas will adopt the Code. Texas has always enjoyed the reputation of being progressive, and it has in this matter an admirable opportunity to enhance that reputation by becoming the first state of the South to adopt boiler-construction laws. It would thereby assume the position of leadership among all the Southern States and its influence would be potent in future endeavors to line these states up in the interest of safety and the conservation of human life.—"Morning News," Dallas, Tex.

Convention of Canadian Stationary Engineers

The twenty-seventh annual convention of the Canadian Association of Stationary Engineers was held at Montreal, Canada, during the week commencing July 24, with the Windsor Hotel as headquarters.

The Montreal Technical School was generously donated for the use of the convention. The sessions of the delegates were held in the handsome amphitheater of the building, and the large hall adjoining was neatly decorated and conveniently arranged for the mechanical display of the Canadian Exhibitors' Association. The representation of supplymen this year was the largest in the history of the organization, there being 53 booths occupied. The exhibition was liberally attended during the week notwithstanding the excessive heat. The attendance of engineers, however, was not up to the standard, as many of them are away at the front.

CANADIAN ASSOCIATION OF STATIONARY ENGINEERS AT MONTREAL

On Tuesday morning at 10 o'clock the convention was formally opened in the amphitheater. Robert Dyson, of Guelph, executive president of the C. A. S. E., occupied the chair and after cordially greeting the delegates, introduced Alderman McDonald, of Montreal, who represented the mayor. Mr. McDonald welcomed the delegates and spoke of the beauties and industrial interests of the city and in conclusion paid a glowing tribute to the preamble of the C. A. S. E. The chairman then presented D. M. Medcalf, chief inspector of Steam Boilers of the Province of Ontario. He considered the Ontario Steam Boilers Act and the general regulations thereto as being as nearly perfect as possible at the present time and hoped that the Province of Quebec would adopt them in their

entirety. He assured the Quebec folks that he would be ready at all times to advise and render any desired assistance. Mr. Medcalf requested the delegates and visitors to carefully inspect the display of officially approved forms, blueprints, surveyors' calculations and other documents and literature of the Ontario Steam Boiler Department, which occupied a table in the center of the exhibit hall. "Dad" Wickens, father of the organization, then addressed the meeting, after which the delegates went into executive session. The necessary committees were appointed, and an adjournment was taken until Wednesday morning at 10 o'clock.

The large amount of business disposed of was due to the general harmony that prevailed among the delegates. The reading of the treasurer's report showed the association to be financially sound.

On Tuesday evening special cars conveyed the convention over the entire mountain route and thence to Highland Park, returning home about midnight.

On Wednesday afternoon at 1:30 o'clock a trip was made to St. Anne's by special cars, returning by steam through Lake St. Louis and shooting the celebrated Lachine Rapids.

On Thursday morning the company enjoyed the hospitality of the Frontenac Brewery, under the guidance of its chief engineer, "Bob" Smith. After a thorough inspection of the plant, good things to eat and drink were plentifully served.

On Thursday evening an entertainment was given in the amphitheater and was thoroughly enjoyed.

At the final meeting of the delegates on Thursday morning the following executive officers were elected: Robert Dyson, Guelph, past president; W. G. Forbes, Montreal, president; F. Gofton, Berlin, vice-president; John Hale, Hamilton, secretary; A. W. Heath, Hamilton, treasurer; John Cooke, Belleville, conductor; John W. Beasley, Chatham, doorkeeper.

The Canadian Exhibitors elected the following officers for the ensuing year: Earl F. Heatherington, president; Lloyd M. Archibald, first vice-president; Charles N. Schrag, second vice-president; B. G. Newton, treasurer; Gordon C. Keith, secretary; L. H. Rummage, assistant secretary; James E. Fiddes, director of exhibits; Alexander W. Crouch, chairman of entertainment committee; Edward L. Foley, chairman of reception committee.

Past President Dyson received a past president's jewel, Brother Sweet making the presentation speech. Three members of the local committee were presented umbrellas in appreciation of good work.

Toronto was chosen as the next meeting place in July, 1917.

PERSONALS

A. D. Quackenbush, formerly of the engineering department of H. M. Byllesby & Co., of Chicago, and for the past few years assistant superintendent of the Mobile Electric Co., has been appointed general superintendent of the latter company.

Harry J. Harp, formerly chief engineer for the Salem Glass Works, Salem, N. J., is now with the Gas Appliance Co., Philadelphia, Penn., as an expert on temperature control and is in charge of the applications to glass lehrs and tanks, tempering furnaces, japanning ovens, etc., of oil and gas burners.

MISCELLANEOUS NEWS

Small Rate Reduction in St. Louis—A reduction of ½c. per kw.-hr. has been announced in the retail rates of the Union Light and Power Co. of St. Louis. Under the new schedule the maximum charge will be 8½c. per kw.-hr. instead of 9c.

The City Council of Mt. Sterling, Ky., failing to obtain what it believes is a just contract from the Kentucky Utilities Co. as to rates and character of equipment for street lighting, will put a $30,000 bond issue for construction of a municipal plant before the people.

The Dakota Power Co. has been granted a permit to build a power dam at Pactola, Minn., in Rapid Cañon, to develop 5,000 hp. Because of the lay of the land, the dam will be 130 ft. high, 300 ft. at the bottom and 650 ft. at the top. It will be of solid concrete, and with a secondary reservoir, the cost is estimated at $1,000,000. Work is to start at once.

The City of Owensboro, Ky., is considering an ordinance which would reduce the cost of electric current for power to 2¾c. per kw.-hr. to manufacturers. Heretofore a special rate of 3¾c. has been made by the municipal plant to manufacturers. This lower rate will be given to manufacturers only and will not be available to them during the peak hours.

U. S. S. "Tennessee" to Be Propelled by Electricity—The contract for furnishing the necessary equipment for the electric propulsion of the superdreadnought "Tennessee," now under construction, has been awarded to the Westinghouse Electric and Manufacturing Co., of East Pittsburgh, Penn. The system of propulsion being installed differs from any now in use by battleships. Instead of the propellers being mechanically connected to the driving engines or turbines, there are two steam turbines developing over 33,000 hp. driving electric generators which furnish current to four 6,700-hp. motors, each motor driving a propeller. In addition to the main generating equipment and propelling motors, the contract includes the auxiliaries for the main turbine-generator sets and smaller auxiliary turbine generators supplying light and power throughout the ship. In all, about 37,500 hp. will be needed.

ENGINEERING AFFAIRS

The Association of Edison Illuminating Companies will hold its annual convention at Hot Springs, Va., Sept. 4 to 7. Headquarters will be at "The Homestead."

The Providence Engineering Society will occupy its new rooms at 29 Waterman St., Providence, R. I., when it opens up for its fall and winter work next September. The whole second floor of the building has been taken over and will be remodeled and equipped in a modern and attractive manner. The society expects to pursue a very interesting and useful line of work for the coming year.

TRADE PUBLICATIONS

C & C Electric & Mfg. Co., Garwood, N. J. Bulletin No. 102-X. Type "IB" motors with commutating poles. Illustrated, 8 pp., 8x10½ in.

Chicago Pneumatic Tool Co., Fisher Building, Chicago, Ill. Bulletin No. E-38. Duntley universal electric hammer drill. Illustrated, 4 pp., 6x9 in.

Allis-Chalmers Mfg. Co., Milwaukee, Wis. Bulletin No. 1632. Centrifugal pumps and centrifugal pumping units. Illustrated, 44 pp., 8x10½ in.

Bayer Steam Soot Blower Co., St. Louis, Mo. Bulletin No. 135. Bayer steam soot blower for horizontal fire tube boilers. Illustrated, 16 pp., 6x9 in.

The Terry Steam Turbine Co., Hartford, Conn., has issued a booklet describing a combination of auxiliary machinery called the Duplex Exciter Set.

Armstrong Cork & Insulation Co., Pittsburgh, Penn. Pamphlet. "Nonpareil High Pressure Covering for Heated Surfaces." Illustrated, 84 pp., 6x9 in.

Hauck Mfg. Co., 140 Livingston St., Brooklyn, N. Y. Bulletin No. 65. Oil burner and torches for emergency and every day service. Illustrated, 8 pp., 6x9 in.

Ohio Brass Co., Mansfield, Ohio. Catalog No. 16. Overhead materials, rail bonds, car equipment specialties, high tension porcelain insulators. Illustrated, 654 pp., 6x9 in.

The Graton & Knight Manufacturing Co., Worcester, Mass. Catalog No. 6. Leather, leather belting, packings, belt cement, belt dressing, etc. Illustrated, 120 pp., 6x9 in.

Link-Belt Co., Chicago, Ill. Bulletin No. 229. Link-Belt traveling water-intake screens. Illustrated, 4 pp., 6x9 in Book No. 249. Insuring the Coal Supply. Illustrated, 24 pp., 6x9 in.

Fairbanks, Morse & Co., Chicago, Ill. Bulletin H178B. Type Y (Semi-Diesel) oil engine. Illustrated, 4 pp., 6x9 in. Bulletin H192C. Type Y oil engines, Style V. Illustrated, 8 pp., 6x9 in.

A. S. Cameron Steam Pump Works, 11 Broadway, New York. Bulletin No. 154. Centrifugal pumps. Illustrated, 16 pp., 6x9 in. Bulletin No. 100. Duplex pumps. Illustrated, 36 pp., 6x9 in.

D. J. Kelsey, 51 Elm St., New Haven, Conn. Circulars. Tilting drafting table with lifting swing blade attachment, draftsman's universal square and protractor, combination angle and scale. Illustrated.

Ingersoll-Rand Co., 11 Broadway, New York. Form No. 3029. Ingersoll-Rogler Class ORC duplex Corliss steam driven air compressors. Illustrated, 32 pp., 6x9 in. Form No. 3036. Turbo blowers. Illustrated, 10 pp., 6x9 in.

Chicago Pneumatic Tool Co., Fisher Building, Chicago, Ill. Bulletin No. 34-Q. A Few Applications of "Giant" Gas and Fuel Oil Engines. Illustrated, 28 pp., 6x9 in. Bulletin E-39. Duntley electric grinders. Illustrated, 8 pp., 6x9 in.

Sprague Electric Works, 527-531 West 34th St., New York. Bulletin No. 48,907. 500-lb. electric hoists. Illustrated, 8 pp., 8x10½ in. Bulletin No. 49,600. Flexible steel armored conductors, flexible steel conduit, stamped steel boxes, fittings and tools. Illustrated, 24 pp., 8x10½ in.